NASA Glenn Research Center

Reduced-Noise Gas Flow Design Guide, Revised

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Also, thanks are due to those who have worked and are working at NASA Glenn Research Center in the field of aircraft noise control and reduction. They have created an impressive array of noise control concepts and technology, some of which have been made use of in this Guide.

Gratitude is expressed to many persons in the noise control field who assisted in locating and evaluating references and effective analyses, and participated in valuable discussions that helped shape this Guide.

Finally, the author wishes to acknowledge the efforts of Beth A. Cooper, Acoustical Engineer and Manager of Noise Programs at Glenn Research Center, for coordination, advice and review of this material. Her vision and leadership in hearing conservation have built a lasting foundation for a quieter future at Glenn Research Center.

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1 April, 1999

1. INTRODUCTION

This "Reduced-Noise Gas Flow Design Guide" (*Design Guide*) is intended as a tool for designers and engineers to facilitate design of gas flow equipment to meet NASA Glenn Research Center (GRC) Hearing Conservation requirements. It provides design guidance and noise emission estimates for native-design gas flow systems. Noise emission estimates are also provided for some mechanical equipment that might be purchased from a vendor[i], but which strongly influence the noise emission of a gas flow system.

This *Design Guide* consists of two parts: the written Manual and a Microsoft Excel® Workbook which implements the noise emission estimates.

This *Design Guide* is to be used in conjunction with the NASA GRC *"Guide for Specifying Equipment Noise Emission Levels"* (*Specifications Guide*)[1], which yields noise emission targets for equipment under particular operational and siting conditions. Although the *Specifications Guide* is directed primarily towards specifying noise emission limits for equipment purchased from vendors, it is used here to provide guidance for native-design gas flow equipment.

The Guide is also to be used in conjunction with the NASA GRC Safety Manual, the Environmental Programs Manual and other applicable regulations.

1.1. Scope

1.1.1. Included in Scope

The Scope of the *Design Guide* includes gas-flow noise that originates in turbulent flow processes within the gas itself and then radiates from piping or vessel walls and from atmospheric vents and other openings. The following gas-flow processes are addressed in the *Design Guide*:

- **Vents to atmosphere**: Gas and steam discharge vents, ambient air intake vents, inlet debris screens,
- **Gas-moving equipment**: compressors, exhausters, fans and blowers,
- **Turbomachinery**: Inlet fan and compressor, combustor core, turbine, exhaust jet mixing and exhaust jet shock cells,
- **Flow noise:** from pipe walls and at fittings,
- **Control valves**
- **Flow measurement devices**: orifices and venturis.

[i] Where available, manufacturers' noise emission or noise isolation data is preferred to estimates computed according to this Guide, although the latter may be used as a "reality check" on the former.

The *Design Guide* also addresses noise control performance of elements typically associated with gas flow systems:

- **Walls**: of pipe, duct and vessels
- **Silencers:** vent silencers and in-line silencers,
- **Acoustical Lagging**

1.1.2. Excluded from Scope

The *Design Guide* relates to Hearing Conservation goals in an industrial environment. Personal comfort issues (including speech intelligibility) related to buildings and office environments are not covered because of their differing requirements[ii].

Although a building HVAC system bears a strong resemblance to the gas flow systems treated in the *Design Guide*, a full treatment of HVAC noise is beyond the scope of this document. For engineering information on these systems, consult Schaffer[2] and ASHRAE[3].

Explicitly excluded from the Scope are mechanical or electrical equipment items (e.g., electric motors, pumps, gears) that do not participate directly in the gas flow. Vendors typically supply such devices. Noise emission limits should be specified according to the *Specifications Guide*[1].

The distribution of sound in rooms, the cumulative effect of multiple noise sources and the benefit of sound absorbing materials are handled in a general way in the *Design Guide*. A detailed study of these subjects is beyond the scope of this document. For more guidance in this area, refer to a good noise control engineering text such as Beranek[4], Bies and Hansen[5], Beranek and Vér[6], or NASA[7].

1.2. Relationship of *Design Guide* to *Specifications Guide*

This *Design Guide* and *Specifications Guide* complement one another. The *Design Guide* provides guidance for noise estimation and reduced-noise design. The *Specifications Guide* was developed under a separate contract to define maximum permissible sound power level (PWL) and/or Sound Level (MPSL) that meet NASA

[ii] Note that levels meeting hearing conservation goals are not necessarily "quiet"; e.g., they do not correspond to a comfortable, office-like environment. In addition, other more stringent noise emission requirements may apply as a result of safety and communications issues.

Glenn Hearing Conservation Goals, and to create a concise specification for purchased equipment that maximizes the likelihood of meeting the specified criterion.

The *Specifications Guide* provides noise emission criteria for individual gas flow system components. Special noise transmission problems arise however because of the interconnected nature of these systems. For example, noise generated by a compressor (provided by Vendor A) may exceed the *Specifications Guide* criterion for noise radiating from piping (provided by Vendor B) at some remote location. While the *Specifications Guide* criterion is still valid, the specification of noise emission becomes more complex in such a case.

The *Specifications Guide* also does not provide guidance for NASA Glenn designers and engineers who seek to design gas-flow system equipment to meet the specified limits.

This *Design Guide* specifically addresses estimation of noise emission for individual components as well as complete gas flow systems for comparison with criteria developed under the *Specifications Guide*.

1.3. Technical Approach of *Design Guide*

This *Design Guide* proceeds from the premise that reduced noise emission can be designed into a system as one of many important performance parameters, and addresses the need for a comprehensive set of design tools related to noise emission.

This *Design Guide* also expresses a strong preference for noise control at the source through good design practice rather than using noise control enclosures, barriers and other noise control elements that can interfere with operational and maintenance goals and space limitations.

Methods are provided for estimating and reducing noise emission at an early design stage to facilitate acceptable noise emission. When it appears that desired noise emission levels cannot be attained, the noise emission estimates facilitate the specification, design and selection of noise control elements provided by vendors. In such cases, more detailed guidance is also available from the Noise Exposure Management Program (NEMP, extension 3-3950).

1.4. Intended Audience

The intended audience for the Guide is designers and engineers with a high degree of technical skill. The user need not have formal training in acoustics, but some degree of familiarity with acoustical concepts such as frequency, sound pressure level, octave- and A-weighted filtering, etc. is presumed. A good overview of the subject matter is available in the NASA "Handbook for Industrial Noise Control"[7].

Engineering texts include Beranek[4], Bies and Hansen[5], and Beranek and Vér[6]. An audio-CD has also been produced for NASA GRC that provides demonstrations of acoustic concepts, as well as auditory and hearing conservation effects[8].

Noise estimation equations provided are in algebraic closed form and do not rely on empirical factors that would have to be derived from acoustical experiments. Parameters of the predictive equations consist of readily available design information, such as mass flow rates, gas properties, pipe diameter and wall thickness, etc. No hand calculations are necessary: the accompanying Workbook (described below) implements the engineering equations described in this text.

Other engineering information is communicated using tables, graphs, diagrams, and sketches. Words defined in the "Definition of Noise Control Terms" (Appendix B, page B-1) are set in italics.

1.5. Feasibility

NASA GRC Hearing Conservation policy requires that equipment noise emissions conform to emission limits derived according to the *Specifications Guide*. Engineering measures to achieve the appropriate levels are often technically feasible but may not be reasonable because of performance, economic or space limitation factors. The Noise Exposure Management Program of the Environmental Management Office should be consulted if the required level of noise control proves to be infeasible in a particular application.

1.6. Support Software

The *Design Guide* is accompanied by a diskette with the following computer software items:

- a Microsoft Excel® workbook (Workbook) entitled "Gas Flow Noise Estimation.xls" that performs noise estimation computations for individual equipment items.
- a Microsoft Word® file entitled "Lagging Specifications.doc" containing a basic acoustical lagging specification identical to that provided in Section 8.2.1 (page 8-4). This document is the same as provided with the *Specifications Guide*.
- a Microsoft Excel® workbook entitled MNEW-1.XLS (for Machinery Noise Emission Worksheet) that assists in the determination of Maximum Permissible Sound Levels for equipment. This workbook is the same as provided with the *Specifications Guide*.

- a Microsoft Word® file entitled "Speclang.doc" that incorporates specification language recommended in the *Specifications Guide*. This file is the same as provided with the *Specifications Guide*.

1.7. Disclaimer

Noise control design of gas flow systems can be an extremely complex engineering task. The *Design Guide* is not intended as a substitute for the services of an experienced noise control professional.

The noise emission estimates reported herein are drawn from the open noise control literature and are believed to be appropriate for the types of gas flow systems present at NASA Glenn Research Center. It should be noted however that they incorporate a number of assumptions that may not apply in particular cases. Therefore, Nelson Acoustical Engineering, Inc. makes no warranty concerning the applicability or accuracy of noise emission estimates produced in accordance with this *Design Guide*.

Finally, the *Design Guide* makes no effort to be original in its methods. Most of the methods recommended in the *Design Guide* are the well-accepted work of others in the noise control field. The methods were selected for appropriate balance between simplicity and accuracy. Every effort has been made to give proper attribution to those whose work has become a part of the Guide. Apologies are offered to any who feel they have been overlooked.

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[1] David A. Nelson, *Guide to Specifying Equipment Noise Emission Levels,* Hoover & Keith, Inc. under contract to NASA Glenn Research Center, 1996. This Guide may be obtained from the Noise Exposure Management Program ((216) 433-3950, or via http://www-osma.grc.nasa.gov/oep/nmtpages/oep_nt.htm)

[2] Mark E. Schaffer, *A Practical Guide to Noise and Vibration Control for HVAC Systems*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1991

[3] *1991 Applications Handbook,* Chapter 42: Sound and Vibration Control, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1991

[4] Leo L. Beranek, Ed., *Noise and Vibration Control, Revised Edition*, Institute of Noise Control Engineering, Poughkeepsie, NY, 1988

[5] David A. Bies and Colin H. Hansen, , *Engineering Noise Control, Theory and Practice, Second Edition*, E&FN Spon, London, 1996

[6] Leo L. Beranek and István L. Vér, *Noise and Vibration Control Engineering, Principles and Applications*, John Wiley & Sons, Inc., New York, 1992

[7] The Bionetics Corporation, *Handbook for Industrial Noise Control,* NASA SP-5108, 1981

[8] David A. Nelson and J. Ashton Taylor, *Auditory Demonstrations in Acoustics and Hearing Conservation*, Hoover & Keith Inc. under contract to NASA Glenn Research Center, 1997

2. USE OF *DESIGN GUIDE*

The goal of this *Design Guide* is to use standard gas-flow system design parameters to obtain noise emission estimates. These noise emission estimates are compared to criteria recommended by the *Specifications Guide* in order to determine if a design is sufficient or if further noise reduction efforts are warranted.

Some of the components covered by the *Design Guide* are mechanical equipment items that might be purchased from vendors. In this case, the noise emission estimates are particularly helpful for old equipment for which acoustical data is no longer available, for new equipment for which data has not yet been developed or not yet obtained from the manufacturer, or simply to provide a check on manufacturers' estimates.

Before beginning to use the *Design Guide* in support of a particular project, take time to consider which equipment is likely to produce significant amounts of noise. Obvious candidates include equipment with a history of or reputation for noisy operation, and large equipment items about whose noise emission characteristics little is known. Consider also the variety of paths that sound might take within the system: if there's a way for the sound to escape into the environment, expect that it will do so at the least favorable location.

Next, use the *Specifications Guide* to develop noise emission criteria for each individual system component. *Specifications Guide* criteria are advantageous because they are flexible enough to allow for a variety of siting and operational considerations. They also provide for a consistent set of criteria amongst equipment designed at NASA and equipment supplied by vendors. A system made up of components specified according to the *Specifications Guide* is expected to be compatible with NASA GRC hearing conservation goals.

Locate the sections of this *Design Guide* Manual that relate to each piece of equipment under consideration. Review the information describing how noise is generated in each case. Awareness of the noise generation mechanisms will help the designer avoid design practices that may be inherently noisy.

Guidelines are provided for reduced-noise design. Consider how these guidelines can be incorporated along with other design considerations. While in some cases these guidelines will complicate the already difficult task of balancing competing design requirements, realize that in many cases implementation of noise control design can actually lead to improved system performance through reduction of turbulence levels, vibration and pressure drop.

Once the design is underway and sufficient information is available, use the Excel® "Gas Flow Noise Esimation.XLS" workbook (Workbook) to estimate the noise emission for each component. For those who are interested and for those who have a unique application that departs from the applications used here, equations describing the noise emission estimates are provided.

Required input parameters are tabulated in the *Design Guide* Manual for each equipment type. The input parameters consist of design parameters commonly available in early project design phases, and do not require performance of any acoustical tests. Note that the Calculator spreadsheet (Section 2.4.2, page 2-6) within the Workbook can be used to estimate unknown required gas flow parameters from known ones.

Compare the estimated noise emission levels to the criterion levels recommended by the *Specifications Guide*. Identify frequency ranges that must be addressed first.

The design of a component may need to be improved iteratively. In some cases it may not be possible to achieve acceptable levels without the assistance of noise control equipment such as enclosures, silencers, and lagging. The noise emission estimates should be helpful in the proper selection of such equipment.

At any point along the way, the results of noise estimates on individual components may be incorporated into a System Analysis supported in the Workbook. Two spreadsheets within the Workbook perform all of the tedious accounting work necessary to estimate the system-related effects.

2.1. Equipment Covered

The gas flow system components listed below are covered by the *Design Guide* Manual and Workbook. They are organized here by principle mechanism of noise generation.

- Free Jets (Chapter 4):

  High velocity gas or steam discharge to atmosphere

- Constrained Jets (Chapter 5):

  Control valves
  Measurement orifices and venturis
  High velocity vacuum intake

- Gas-Moving Equipment and Flow Interaction with Structures (Chapter 6):

  Compressors and exhausters
  Fans and blowers
  Flow noise from pipe walls and fittings
  Air inlet debris screen

- Turbomachinery (Chapter 7):

  Inlet fan and compressor
  Combustor
  Turbine
  Jet exhaust (mixing and shock-associated noise)

- Noise Controlling Elements (Chapter 8):

  Pipe, duct, vessel and tank walls
  Silencers
  Lagging
  Reflection of noise at an open pipe end

2.2. *Specifications Guide* Criteria

The *Specifications Guide* requires that equipment noise levels not exceed a maximum permissible sound level (MPSL) when measured under appropriate load 1 meter from the equipment. For equipment sited outdoors, limiting octave band sound power levels are also given.

A baseline noise emission criterion (in A-weighted dB re 20 Pa**)** is assigned to each Equipment Group defined in the *Specifications Guide*. The MPSL may differ from the baseline noise emission criterion, depending on seven adjustments that take into account various siting and operational characteristics. The net adjustment may be between –10 dB(A) and + 25 dB(A). Baseline noise emission criteria for the equipment types covered in this *Design Guide* are listed below according to Group numbers assigned in the *Specifications Guide*:

- **Group 1: Heavy Machinery**

  Control valves
  Measurement venturis and orifices
  Compressors and exhausters
  Blowers and fans

- **Group 2: Vents to Atmosphere**

  High velocity discharge of gas or steam to atmosphere
  High velocity intake of air from atmosphere
  Air inlet debris screen

- **Group 3: Piping and Ductwork**

  Flow noise generated at pipe walls and fittings

- **Group 4: Light Machinery**

  Building ventilation fans or blowers

- **Group 5: Transformers** (Not applicable to this *Design Guide*)

2.3. Equipment Types Covered

Equipment types covered by the *Design Guide* are arranged into four general classes depending on the primary method of noise generation. One Section is devoted to each:

- Free Jets (Chapter 1)
- Constrained Jets (Chapter 5)
- Gas-Moving Equipment and Flow Interaction with Structures (Chapter 6)
- Turbomachinery (Chapter 7)

Section 7 deals with turbomachinery noise from an industrial noise control standpoint. NASA GRC has produced a large body of research on aircraft engine noise and noise control over the years. Some of that research has been incorporated into the *Design Guide*. The equations are used in simplified form to predict gross behavior.

Section 8 addresses elements that reduce or constrain noise, including pipe walls, silencers, acoustical lagging, and rooms.

For each case, four types of information are provided:

- The physical mechanisms of noise production, noise reduction and noise transmission explained for each class of equipment.
- Design guidelines for reduced-noise equipment operation.
- Design parameters required for noise emission estimation using the Workbook and notes on their use are enclosed in a box.
- Predictive equations for noise emission, based on readily available design information.

2.4. Workbook

A Microsoft Excel® workbook (Workbook) has been developed to accompany the *Design Guide*. The Workbook comprises a series of spreadsheets that perform the noise emission estimation and noise control calculations.

The workbook spreadsheets and most cells are protected and the file is saved in read-only format to prevent accidental erasure or modification. To modify the workbook or access its contents, it may be unlocked using the sequence Tools / Protection / Unprotect Sheet. No password is required.

Each spreadsheet presents a computation form that guides the user through entering the relevant input parameters. Data entry is made in unlocked cells denoted by white background, bold type and a black outline. Units for data entry are selectable by the User by means of drop-down lists. Units may be mixed without restriction.

A blue background denotes spreadsheet outputs. Units for outputs are selectable by the User by means of drop-down lists. Units may be mixed without restriction.

Noise emission estimates are compared directly with a Maximum Permissible Sound Level (MPSL) value entered by the User and with the sound power level limits for outdoor equipment. To assist the User in determining which octave bands must be reduced to achieve the criterion, octave bands that individually contain enough energy to exceed the criterion are denoted by a bright red background and bold, white characters. An orange background with bold, black characters denotes octave bands that individually are within 5 dB of the criterion. If the criterion is exceeded, further noise reduction must be obtained by working on the octave bands with red cells in them and may require to octave bands with orange cells as well.

For comparison with the octave band sound power level criterion, a bright red background and bold, white characters denote octave bands that exceed the criterion. An orange background with bold, black characters denotes octave bands that individually are within 5 dB of the criterion. Note that these color codes are provided for information only: when the criterion is exceeded, additional noise control must begin in the bands with red cells and may be required in the bands with orange cells as well.

Octave-band values intended for use as inputs to the System Input-Output spreadsheet are highlighted with a salmon-colored background. A light yellow background denotes tabular information. Octave-band sound power level criteria are displayed with a gray background.

Examples of each of these formats along with other helpful information can be found in the "Read Me" spreadsheet of the Workbook.

2.4.1. Single Component Design

When designing a single component, each Spreadsheet may be used to provide a "stand-alone" estimate of the radiated and in-duct sound power level, as well as the Sound Pressure Level at a location of the User's choice. Octave band and A-weighted output values are provided.

Some noise control devices have noise control equipment that is an integral part of the device, e.g., in-line silencers for valves. In such cases, a calculation of their noise control benefit takes place directly on the spreadsheet for that device.

Spreadsheets are provided for the following equipment types:

- Intake Vents
- Venturis and Orifices

- Inlet Debris Screen
- Control Valves
- Compressors and Exhausters
- Blowers and Fans
- Jet Engine Fan and Inlet Compressor
- Jet Engine Combustor
- Jet Engine Turbine
- Jet Exhaust Mixing and Shock-Associated Noise
- Gas Vents and Reliefs
- Steam Vents and Reliefs
- Flow Noise in Pipes
- Preliminary Silencer Selection
- Pipe and Duct Wall Transmission Loss
- Reflection Loss at Pipe End with Flow
- Gas Flow Calculator

2.4.2. Gas Flow Calculator

A spreadsheet is included that serves as a general calculator useful for gas flows and noise emission. It facilitates conversion of known parameters into required inputs when these are unknown. The spreadsheet includes calculators for the following:

- *Ideal Gas*: solve for Pressure, Temperature or Density given the other two.
- *Isentropic Expansion and Contraction*: solve for Temperature, Density, Velocity and Sonic Velocity of an expanded or contracted gas from the pressures before and after expansion or contraction.
- *Velocity, Mass Flow and Volume Flow Conversions:* Find any two of the three given the other and pipe diameter.
- *Sonic Velocity and Mach Number:* from Gas Velocity and Temperature
- *Units Converter:* Convert values from one system of units to another
- *Decibel Mathematics:* addition and subtraction of decibel spectra, and three types of wave divergence computation.

3. NOISE EMISSION AND CONTROL

The key to reducing noise of gas flows is to understand the mechanisms by which noise is produced and transmitted to the environment, and to design for the opposite result to whatever extent possible.

3.1. General Discussion

Noise is a waste byproduct of mechanical processes. A very small fraction of the mechanical energy in a given process reaches our ears as sound. The fraction is small primarily because of various inefficiencies in converting mechanical energy into acoustic energy.

For most mechanical equipment, casing vibration creates waves radiating into the atmosphere with an efficiency ranging from 10^{-5} to 10^{-7}. In other words, it may take as much as one megawatt of mechanical power to produce one acoustic watt. While that may at first seem encouraging, one acoustic watt is a rather large quantity that is capable of causing hearing damage to personnel nearby.

Gas flow systems are potentially more noisy than mechanical equipment, however, because the mechanical power is already part of the gas flow: there is considerably less mechanical/acoustical conversion inefficiency to overcome. Gas flow systems convert their mechanical power to acoustical power at efficiency rates ranging from 10^{-3} to 10^{-5}. This is especially problematic when the gas flow is not contained within piping but comes into direct contact with the atmosphere. High velocity gas discharge vents and aircraft engine exhausts are cases in point.

For perspective, it is worth noting that loudspeakers and other similar devices specifically designed to radiate sound do so with efficiency of approximately 10^{-2}, or 1%.

Let us summarize the above using W_M for stream mechanical power at the point of noise generation, W_A for acoustic power and η for efficiency, and substitute for W_M:

$$W_A = \eta W_M$$
$$W_A = \eta FV$$

Assuming that the force acts over the same area the flow passes through, this can be simplified further to

$$W_A = \eta \Delta P Q$$

It should be clear from this simplified approach that in order to reduce noise output, three primary options are available:

- Reduce efficiency of conversion to acoustic power,
- Reduce force exerted on the gas by reducing either the pressure differential or the area over which it acts,

- Reduce the velocity of the gas by reducing the volume flow or increasing the flow area.

A fourth important option is not obvious from the above list:

- Modify the design to cause energy to be expressed frequency bands less likely to cause hearing damage.

Judicious application of these four approaches is the key to successful noise reduction in gas flows.

3.2. Sound Power Level and Sound Pressure Level

It is important to properly understand the distinction between sound power and sound pressure. The acoustic power of a source in watts is called the *sound power*. This quantity represents the energy output of the source per unit time into its environment. *Sound power level* is a decibel expression of the sound power referenced to 10^{-12} watts:

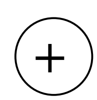
$$L_W = 10 \log_{10} \frac{W_A}{10^{-12}\, watt}$$

Sound pressure is the expression of that energy filling the environment, just as temperature is the expression of thermal energy filling the environment. In the case of heat, it is clear that the temperature in a heated space is a function not only of the power of the heater, but also on the proximity of the observer, the ability of the environment to contain heat, and the ambient temperature that would prevail independent of the heater.

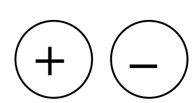
3.3. Noise Generation in Gas Flows

Three types of acoustic sources are responsible for most of the noise in gas flows: monopoles, dipoles and quadrupoles.

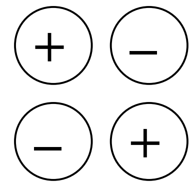
A *Monopole* is the simplest type of source, corresponding to a pulsation of gas pressure or velocity. A monopole source is like a pulsating sphere. Pressure or velocity pulsations are in phase at all points on the source. A vibrating duct wall or open end of a pipe might serve as a monopole under certain conditions.



Dipoles are the consequence of oscillatory forces in the flow arising chiefly from interactions between the gas and structures. A dipole is analogous to two monopole sources oscillating out of phase and separated by a small distance



(compared to a wavelength). Compressor blades give rise to dipoles because they have high pressure on one side and low pressure on the other. Dipoles also are found in the periodically alternating vortices shed from flow obstructions such as struts. In many cases, dipoles lead to tonal (pitched) components in the noise.



Quadrupoles result from viscous stresses in turbulent flow in the absence of obstacles. A quadrupole is analogous to two dipoles that oscillate out of phase and separated by a small distance (compared to a wavelength). Wherever turbulence and/or mean velocity gradient are high, quadrupole source strength may be significant.

For a given mechanical power and size compared to an acoustic wavelength, a monopole is the most efficient radiator of sound, followed by dipoles and quadrupoles. The amount of sound energy radiated, however, is proportional to u^4, u^6 and u^8, respectively, where u is a local flow perturbation (acoustic) velocity. Thus at high velocities (on the order of Mach 1) the quadrupole source strength can predominate over dipoles and monopoles.

In general, noise control is most effective when all three types of acoustic sources are minimized. However, where one type is predominant, minimizing the conditions that give rise to that particular type of acoustic source is usually the most successful approach.

3.4. Noise Emission

Noise is emitted from the gas flow system in one of three ways:

- by radiation from a gas flow boundary where the noise is produced, such as for a high velocity unconstrained gas jet.
- by radiation from pipe and duct walls, which vibrate in response to fluctuating pressures due to turbulent processes or acoustic excitation within,
- by radiation of noise within the piping system from an intake or discharge opening in the system.

3.4.1. Noise Emission from a Flow Boundary

The magnitude of noise emitted at a gas flow shear boundary depends chiefly on the velocity of the gas jet relative to the ambient atmosphere, but also on the nozzle area and on the density and temperature of the jet relative to the atmosphere. Sound generated within the gas jet core is refracted on passing through the shear layer. More detail is given in Chapter 1 and in the references given.

3.4.2. Noise Emission from Piping

Noise generated within a piping system (e.g., by a compressor) propagates in both the upstream and downstream directions if the flow is subsonic. As the flow nears sonic velocity, most of the energy travels in the downstream direction. Above sonic velocity, all of the sound energy is convected downstream with the flow.

As sound energy travels through the piping, a small fraction is expended in vibrating the pipe or duct walls, which in turn re-radiate the sound to the environment. The remainder of the sound energy usually remains within the gas. This is beneficial on one hand because sound levels outside the piping are reduced. On the other hand, the sound energy trapped within the pipe travels great distances, often without significant attenuation. When the sound energy emerges at remote locations, unintended noise emission problems can arise.

No significant loss of acoustic energy should be expected along the first several hundred diameters of round piping length. The System Analysis model in the Workbook assumes no acoustic loss other than sound transmission along the length of any pipe.

Higher values of pipe- or duct wall *transmission loss* indicate lesser fractions of sound transmitted. It turns out that circular pipe has high transmission loss in all but a small band of frequencies, and sound levels decay only very slowly with distance (a fraction of a dB per 100 diameters). By contrast, rectangular duct profiles have significantly lower transmission loss at low frequencies, and release a greater proportion of sound to the environment. However, in that case the sound levels decay more rapidly with distance.

Methods of reducing noise emission from piping are discussed in Chapter 8.

3.4.3. Transmission from Open Duct End

Sound propagating within a piping system may eventually reach an intake or discharge opening. An abrupt acoustic impedance change at the open end causes some waves (particularly at low frequency) to be unable to exit the opening. This effect is increased by significant inflow and decreased by significant outflow.

Horn-like structures at the end of the duct may actually increase sound radiation by diminishing the impedance change.

3.4.4. Radiation to Environment

Within the *near field* of a source of sound (within approximately one source dimension) the sound pressure level fluctuates considerably but on the whole does not decay with distance. In the far field (several source dimensions distant), the sound pressure level decreases approximately 6 dB per doubling of distance as long as there are no reflecting surfaces (other than the ground) present. At greater distances outdoors, levels may decrease more rapidly because of atmospheric and ground effects. Practical control of the received level outdoors can only be achieved by reducing the level of the source, or by erecting a barrier or enclosure close to either the source or receiving point.

When multiple sources of noise are present, the sound energies produced are additive. In such a case sound pressure levels will generally be higher (by as much as 5 dB(A)) than the highest sound pressure level produced by any one piece of equipment. For this reason, noise control efforts must begin with the equipment producing the highest sound pressure level and can only be expected to reduce levels to those produced by the equipment not treated. For example, suppose two machines each produce 85 dB(A) at a given location. The combined level would be 88 dB(A). If noise control were applied to reduce one source from 85 dB(A) to 65 dB(A), the combined level would be 85 dB(A). Thus, a 20 dB(A) noise control treatment yielded in this case a net benefit of 3 dB(A). (See Appendix C for details on decibel mathematics.)

In an indoor environment, reflected sound tends to build up so that sound levels decay less rapidly with distance, reaching an approximately constant level. Increasing the surface area covered with sound absorbing material can reduce the reverberant level. This is especially important when multiple equipment items are present: the reverberant sound pressure levels from individual equipment items are additive. A reverberant space causes otherwise "local" noise emission challenges to become "global" ones that may effect may locations and employees with a building.

4. REDUCED-NOISE DESIGN FOR FREE JETS

A free jet is defined for the purposes of the *Design Guide* as an unimpeded discharge of high velocity gas into the atmosphere. Free jets include gas and steam discharge to atmosphere. The "jet" in question is of an industrial character, wasting its thrust as it escapes (in most cases) from the open end of a pipe. Where a more formal nozzle is used that is intended to maximize thrust, the discussion of aircraft jet engine mixing and shock-associated noise in Section 7.4, (page 7-3) may be more relevant.

Note that the jet formed by an intake (vacuum) vent is not free but constrained within downstream piping or a vessel. Intake vents are discussed in Section 5.2.3, page 5-5.

4.1. Mechanism of Noise Production for Free Jets

High velocity gas interacts with the surrounding atmosphere at rest to produce significant shear stresses and turbulent mixing. This mixing produces sound. The overall sound power output W_A of the jet is taken to be dependent on the eighth power of exit velocity U_j after Lighthill[9]:

$$W_A \propto \frac{\rho_j S_j U_j^8}{c^5}$$

where

ρ_j is the jet density,
S_j is the fully expanded jet area, and
c is the sonic velocity in ambient air.

Small-scale vortices give rise to high frequency quadrupole sound sources. Larger scale vortices within the jet produce low frequency quadrupole sound sources. The frequency at which peak sound pressure occurs is approximately:

$$f_p = \frac{0.2U_j}{D_j}$$

where

f_p is the peak frequency in Hz, and
D_j is the fully expanded jet diameter.

When the ratio of upstream to ambient pressure P_I/P_A is greater than 1.5, sonic flow may exist in the vena contracta downstream of the outlet. If the ratio exceeds 1.89 (in air), the flow will definitely be sonic ($M_j > 1$). Once sonic flow is reached the flow cannot accelerate further without the help of a converging-diverging nozzle. If no C-D nozzle is present, *choked flow* is said to exist. Any further increase in flow comes about through an increase in density and entropy that resolves in shock waves in the downstream flow. Shock waves are efficient generators of noise and further increase noise emission.

If the exhaust stream is interrupted by any kind of obstacle, noise emission may be increased by as much as 10 dB(A).

Noise radiated from free jet mixing has a pronounced directionality that arises from convection of quadrupoles by the flow and by refraction at the shear boundary. Peak levels are reached 150° from the inlet axis (30° from the discharge axis). Noise emission from shocks is normally taken as omnidirectional.

An empirical model that takes into account the gross behavior of gas and steam jets based on upstream pressure and temperature and nozzle area is given below in Section 4.5 (page 4-5).

4.2. Gas and Steam Discharge

Gas and steam discharges are characterized by high pressure gas venting through a control valve, relief valve, burst disk, or similar opening to atmosphere. Continuous and intermittent vents are included in this definition, as are blowdown applications in which a stationary volume of gas is vented.

The applications here are industrial. Because the vented gas serves no further useful purpose, noise control options that reduce thrust are acceptable. For the case of aircraft engine components, jet exhaust is discussed separately in Section 7.4 (page 7-3).

4.3. Guidelines for Noise Control of Gas and Steam Discharges

Significant noise reduction is possible by use of a vent silencer in conjunction with a properly selected control valve.

- ***Employ a Vent Silencer***: A vent silencer consists of two stages; a diffuser basket and a dissipative silencer. The diffuser basket breaks the jet into a number of small jets, increasing the peak frequency and thus rendering the dissipative silencer more effective. Reductions of 10 db(A) to 50 dB(A) are achievable with various designs. Care should be taken that the self-noise of the silencer does not limit its performance.

- ***Use a Low-noise Valve:*** Noise generated at the control valve also propagates with the flow, but is not frequency-shifted by the diffuser basket. Thus the vent silencer is less effective against valve noise than against jet noise. Reductions of valve noise by vent silencers are on the order of 5 dB(A) to 35 dB(A). The downstream sound power output of the valve should be at least 15 dB(A) less than the unsilenced sound power output of the jet.

  Guidelines for reducing valve noise are given in Section 5.2.

Smaller noise reduction gains can be achieved using these methods:

- ***Reduce turbulence upstream of the exit:*** allow 6-10 pipe diameters of straight duct length before the exit or other impediment is reached. Noise emission can be increased by 5 dB(A) or more if turbulent flow reaches the exit. Note that control valves and support struts are examples of flow impediments that produce turbulence.

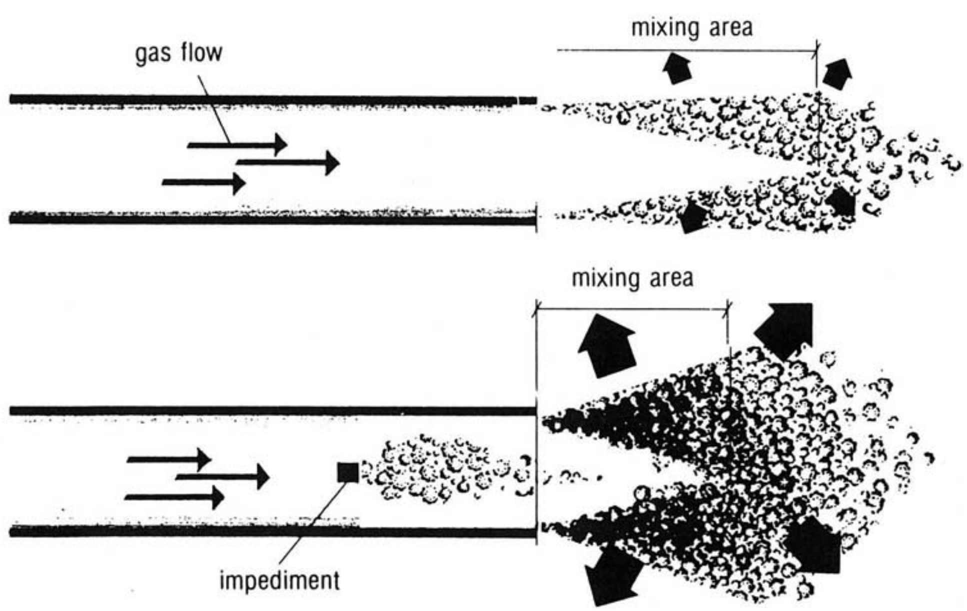


Figure 1: Effect of Turbulence Upstream of Exit

(Ingemansson and Folkesson[10])

- ***Angle of Radiation***: The axis of discharge should be oriented at least 90° away from noise sensitive areas, otherwise there is no benefit. For very large diameter outlets the benefit may be 5 dB(A) or more.

- ***Use Larger-Diameter Piping Downstream of Valve:*** The main reason for taking this action is that the dominant frequency is reduced by about 1 octave (see 4.5.2, page 4-6). The benefit of favorable directional orientation is also greater for a larger opening. In general, the net benefit is on the order of 3 dB(A) to 5 dB(A).

  A special case of this treatment is the can-type supersonic suppressor as developed by NASA GRC[11]. In this treatment, the discharge pipe is deliberately made long enough that the emerging jet boundaries strike the pipe walls. An additional 3 dB(A) to 5 dB(A) reduction may be possible.

  While abrupt area changes in flows are usually not beneficial because of increased turbulence, here the turbulence is increased so dramatically that the flow is decelerated before reaching the exit.

- ***Reduce the Pressure and Temperature of the Vented Gas,*** although this may seldom be practical. A 20% reduction in pressure yields a 1 dB(A) reduction, while a 20% reduction in gas temperature yields a 2 dB(A) noise reduction.

- ***Entrain ambient airflow*** using a co-annular eductor nozzle to reduce relative velocity in the shear layer[11]. Overall reductions of between 5 dB(A) and 10 dB(A) can be achieved using this approach. See Figure 2 below.

- ***Introduce a rotary component to the jet flow*** using radial vane structures. This works best for hot gas exhausts.[12]

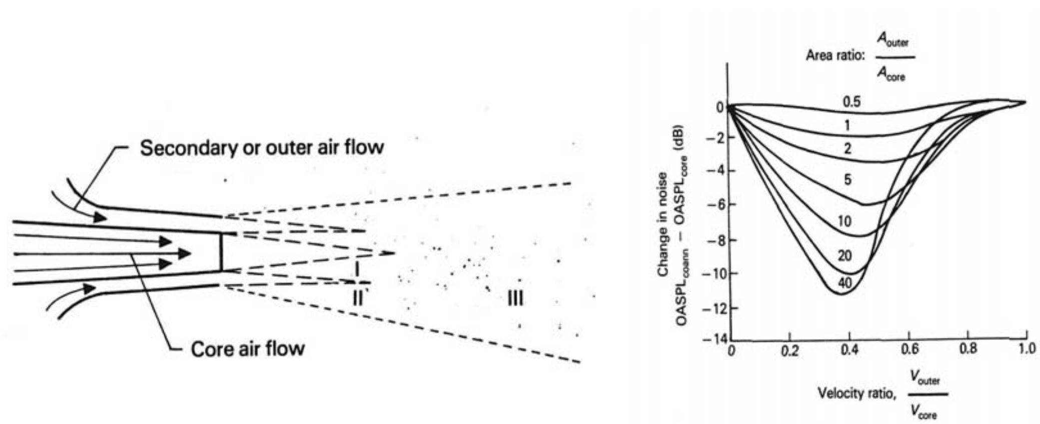


Figure 2: Effect of Entraining Airflow

(after Huff[11])

4.4. Noise Emission Estimation Using Workbook

Spreadsheets:

- Gas Vents
- Steam Vents

Required Inputs:

- Upstream conditions: pressure P_1, temperature T_1, volume V, moisture $\%m$, superheat temperature T_s
- Valve, Piping and Nozzle: valve diameter, D_v, downstream pipe diameter D_D, silencer outer diameter D_o, nozzle coefficient C_N
- Downstream conditions: pressure P_2, temperature T_2,
- Observer: distance r, angle θ

Notes:

- Reservoir volume V is an optional input
- Nozzle coefficient C_N is assumed to be 0.85 unless otherwise known.
- If no silencer is used, set silencer diameter equal to downstream pipe diameter.
- Expanded temperature and expanded density of flow must be determined from steam tables assuming that the gas has reached ambient pressure.

4.5. Predictive Equations for Discharge Vents

The Sound Pressure Level is estimated from factors for the overall sound power level, spectral shape, directivity and geometric spreading with distance.

$$SPL(f, r, \theta) = L_{W,overall} + \Delta\left(\frac{f}{f_p}\right) + D(\theta) + G(r)$$

4.5.1. Overall Sound Power Level

For air and all other gases, the emitted sound power level (dB re 1 pW) is estimated as[13]:

$$L_{W,overall} = 10\log_{10}\left(P_1 A_V C_N\right) + 20\log_{10}\left(\frac{T_1}{G}\right) + 85$$

where P_1 is pressure upstream of the control valve in psia, A_V is the valve open area in square feet, C_N is the nozzle coefficient (assumed to be 0.85 unless otherwise known), T_1 is the upstream temperature in degrees Rankine (°R) and G is the specific gravity of the gas. Downstream conditions are taken to be air at sea level, standard temperature and pressure.

For steam, the emitted sound power level (dB re 1 pW) is estimated as[13]

$$L_{W,overall} = 17\log_{10}\left(51.43 P_1 A_V C_N F_m F_s\right) + 50\log_{10} T_1 - 85$$

$$F_m = \frac{1}{1 - 0.012\,\%m}$$

$$F_s = \frac{1}{1 + 0.00065 T_s}$$

where m is the percentage moisture, and T_s is the number of degrees of superheat (°F) for superheated steam.

The peak frequency of emitted noise is

$$f_p = \frac{0.4 c_j M_j}{D_V}$$

where D_V is the valve throat diameter and c_j is the speed of sound within the gas jet at the valve exit. The speed of sound c_j can be expressed in feet per second as

$$c_j = 223\sqrt{\frac{\gamma T_j}{MW}}\ \text{ft/s}^{-1}$$

$$T_j = \frac{T}{1 + \frac{\gamma - 1}{2} M_j^{\,2}}.$$

4.5.2. Spectral Shape Function Δ

The spectral shape function Δ is tabulated below as a function of the ratio of frequency to the peak frequency f/f_p. The spectral shape corrections convert the overall sound power level L_W give octave band values for two cases, here designated A and B. Case A corresponds to either no downstream piping or downstream piping the same size as the valve. Case B corresponds to downstream piping larger than the valve.

The effect of larger downstream piping after the valve is to shift the peak about one octave down in frequency. The spectral shape changes only slightly.

Table 1: Spectral Correction Factors for Gas and Steam Vents

	Frequency ratio f/f_p												
	1/128	1/64	1/32	1/16	1/8	1/4	1/2	1	2	4	8	16	32
A	*-40*	*-36*	*-30*	*-24*	*-18*	*-12*	*-6*	*-4*	*-6*	*-12*	*-18*	*-24*	*-30*
B	*-40*	*-33*	*-22*	*-15*	*-9*	*-6*	*-5*	*-6*	*-11*	*-19*	*-29*	*-40*	*-50*

The correction factors are approximated by the function:

$$\Delta\big|_A = -4.6 - 1.78x^2 - 3.9681\times10^{-4}x^4$$

$$\Delta\big|_B = -6.3 - 3.40x - 1.59x^2 + 0.1527x^3 + 0.1933x^4 + 2.7025\times10^{-4}x^5 - 1.3718\times10^{-4}x^6$$

where

$$x = \frac{\log\left(f/f_p\right)}{\log(2)}$$

4.5.3. Directivity Factor *D(θ)*

Table 2, Table 3 and Table 4 below give directivity factors for high velocity gas or steam discharge as a function of the diameter of the outlet in inches. These values are added to the sound power level.

Table 2: Gas or Steam Discharge, 0° from Axis

	Octave Band Center Frequency [Hz]								
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
8	0	0	0	0	0	0	0	0	0
15	0	0	0	1	1	1	1	1	1
26	1	1	1	2	2	2	2	2	2
36	2	2	3	3	4	4	4	4	4
54	3	3	4	4	5	5	5	5	5
72	4	4	5	5	6	6	7	7	7

Table 3: Gas or Steam Discharge, 45° from Axis

Octave Band Center Frequency [Hz]									
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
8	0	0	0	0	0	0	0	0	0
15	0	0	0	0	0	0	0	0	0
26	0	0	0	0	1	1	1	1	1
36	0	0	1	1	2	2	2	2	2
54	1	1	2	2	3	3	3	3	3
72	2	2	3	3	4	4	5	5	5

Table 4: Gas or Steam Discharge, ≥90° from Axis

Octave Band Center Frequency [Hz]									
Diam. [in]	31.5	63	125	250	500	1000	2000	4000	8000
4	0	0	0	0	0	0	0	-1	-3
5	0	0	0	0	0	0	-1	-3	-6
8	0	0	0	0	0	-1	-2	-5	-11
15	0	0	0	0	0	-1	-3	-7	-13
26	0	0	0	0	-1	-3	-5	-9	-14
36	0	0	0	-1	-3	-6	-7	-11	-15
54	0	0	-1	-2	-5	-8	-10	-13	-16
72	0	-1	-2	-5	-7	-10	-12	-15	-17

4.5.4. Self Noise

A Vent Silencer (see Section 8.4, page 8-11) reduces the noise of the expanding gas flow by first converting one large jet to a large number of very small ones using a "diffuser basket". The resulting high frequency sound is then effectively absorbed as the gas flow passes between parallel baffles of sound absorbing material.

This process produces additional noise of its own called "self-noise" (Section 8.4.5, page 8-12). The self-noise sound power level is added on an energy basis to the silenced vent sound power level (in dB) to find the residual sound power level at the silencer exit.

[9] M. J. Lighthill, On Sound Generated Aerodynamically, II., Turbulence as a Source of Sound, Proc. Roy. Soc. (London) Ser. A, vol. 222, no. 1148, Feb. 1954

[10] Stig N. P. Ingemansson, Claes Folkesson, *"Noise Control: Principles and Practice"*, this illustration from Noise News International, Vol. 3 No. 3, 1995 Sept., pp. 178-183. Published in book form by the American Society of Safety Engineers as *"Noise Control: A guide for workers and employers"*.

[11] R. H. Huff, A Simple Noise Suppressor Design for Vented High Pressure Gas, NASA Tech. Brief, summer 1979, p. 278.

[12] I. R. Schwartz, Minimization of Jet and Core Noise by Rotation of Flow, NASA Tech Brief B75-10131, 1975

[13] Bill G. Golden, Jim R. Cummins jr., "Silencer Application Handbook" , Universal Silencer, Stoughton, Wisconsin, 1993

5. CONSTRAINED JETS: CONTROL VALVES, ORIFICES, VENTURIS, VACUUM VENTS

A constrained jet is a high velocity discharge of gas into a constrained area, such as a pipe, tank or vessel. Constrained jets exist downstream of control valves, measurement orifices and venturis, and intake vents.

5.1. Mechanism of Noise Production for Constrained Jets

Constrained jets are the result of an in-line flow restriction. At the restriction the flow velocity increases and, from Bernoulli's theorem, it is known that a corresponding pressure reduction occurs. The point of maximum flow velocity and minimum static pressure is called the *vena contracta* and is located a fraction of a restricted diameter downstream.

The boundary between the fast-moving jet and slower moving gas in the pipe is the site of large shear stresses that generate small-scale vortices, with larger scale vortices created within the gas jet. The physics of the gas jet differs little from a free jet until the expanding jet contacts the walls. The difference lies in the interaction of the flow with the walls. Quadrupoles in the shear layer strike the outer wall and, along with their in-phase reflected pairs, create dipoles. The forces exerted on the pipe wall cause it to vibrate and in turn to radiate sound into the surrounding environment. Within the pipe, noise propagates through the gas and is convected with it. As sonic flow is approached, it becomes increasingly difficult for sound to travel upstream. For this reason, noise emission is often concentrated downstream of flow restrictions in control valves and on vacuum inlet vents.

No fluid is completely inviscid, so passage through the restriction incurs a pressure loss equal to $1/2K\rho U^2$ where K is a dimensionless loss factor. From Bernoulli's theorem of isentropic flows, the flow through the restriction can be shown to be:

$$Q = UA = \sqrt{P_1 - P_2} \times \sqrt{\frac{2}{\rho K}} \times A$$

where U is the mean flow velocity through the restriction or area A. The valve sizing coefficient C_V is derived from this expression as

$$C_V = \sqrt{\frac{2}{\rho_W K}} \times A$$

and assigned a numerical value for water flows expressed in gallons per minute and differential pressure in pounds per square inch, such that

$$Q = C_V \sqrt{\Delta P}$$

Note that the value C_V/D_V^2 is a property of the valve at a particular flow condition. Actual valve sizing for real gases is more complex than can be addressed in the

Design Guide. Consult valve catalogs and sizing routines and software from control valve manufacturers.

At sonic flow speeds shock waves form and the flow is no longer isentropic. Catalog values of C_V are intended to account for all of the added complexities of the flow. Furthermore, because control valves can often be used for fluid or gas flow, catalog C_V values are often applied in gas applications. A more thorough treatment of control valve flows is contained in the literature of control valve manufacturers[14,15]

The mechanical energy in the flow is proportional to $Q\Delta P$ or $C_V\Delta P^{1.5}$. The efficiency of noise generation is proportional to the flow velocity at the point of noise generation, that is, within and just downstream of the restriction, and is noticeably increased when shocks form.

The peak flow velocity is attained in the *vena contracta*. The degree to which the *vena contracta* pressure P_0 falls below the downstream pressure P_2 is called *pressure recovery*. The *pressure recovery factor* F_L in common use for control valves is defined as

$$F_L^{\;2} = \frac{P_1 - P_2}{P_1 - P_0}$$

where P_1 is the upstream pressure. The factor F_L takes values between 0 and 1. When F_L is small, pressure recovery is complete and $P_1 - P_0 >> P_2 - P_0$. Because P_0 is less than P_2, the velocity in the *vena contracta* is higher than would be expected from the service pressure drop $P_1 - P_2$. The increased velocity corresponds to increased noise output. When $F_L \approx 1$, $P_2 = P_0$, there is no pressure recovery and the flow velocity in the *vena contracta* is essentially that in the downstream pipe. This situation usually corresponds to minimum noise output for a given pressure drop.

Confusion may result because a "high" value of F_L corresponds to low pressure recovery, and vice versa. The high value is actually preferred, because it minimizes the flow velocity in the *vena contracta* for a given pressure drop. By contrast it should be clear that the pressure drop that causes sonic flow within the restriction (and consequently high noise emission) is smaller when F_L is low than when it is high.

The spectral shape of the noise emitted is similar to that for a free jet, being centered around a peak frequency f_p

$$f_p = \frac{0.2 M_j c_0}{D_j} \quad M_j < \sqrt{2},\ c_0 = \text{sonic velocity in vena contracta}$$

$$f_p = \frac{0.28 M_j c_0}{D_j \sqrt{M_j^2 - 1}} \quad M_j \geq \sqrt{2}$$

The noise radiated through the pipe walls is influenced by the frequency-dependent wall transmission loss (see Section 8.1).

5.2. Guidelines for Reduced-Noise Design

Noise of constrained jets and flow restrictions is reduced using general approaches described below. Noise reduction techniques for control valves are discussed in several references[14,15, 16,17].

Ultimately, these techniques relate to reducing the mechanical energy in the gas, the flow through a restriction, upstream turbulence, and the propagation of generated sound waves along the pipe or through the pipe into the environment.

5.2.1. Control Valves

- ***Multi-port resistance plates*** (also called diffusers) are appropriate for large pressure drops where a small control range is required. The plate should be sized for the maximum flow condition with the control valve 100% open. The control valve is then sized to be 30% or more open at minimum flow. Noise reduction of 15 dB(A) is achievable for a fixed control point. The benefit is reduced for greater departures from the maximum flow condition.
- ***Valve Trim:*** Some forms of **valve trim** provide special flow control elements (e.g., a series of perforated disks) whose purpose it is to provide pressure drop in stages. More gradual deceleration reduces the pressure recovery. Check with manufacturers regarding the availability of valve trim for the control valve in question: it may not be available for all valve types and sizes and is often difficult to install in retrofit situations. An example of valve trim is depicted in Figure 3. Although this particular trim is intended for liquid service, it demonstrates the principles clearly.

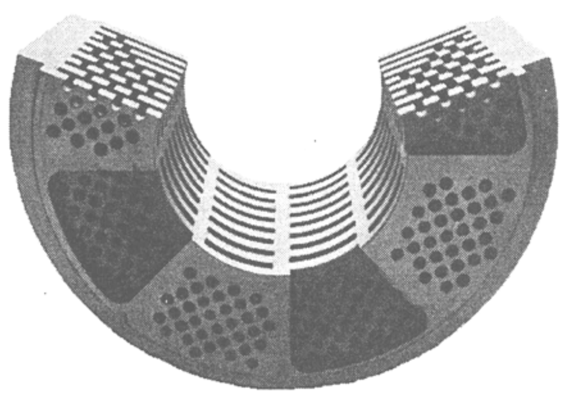


Figure 3: An Example of Valve Trim

(Masoneilan/Dresser)

- ***Apply lagging*** to the exterior of the pipe. Focus on the area downstream of the valve.
- ***Multiple flow paths:*** If control over a wide range is required, consider using multiple control valves mounted in parallel, each sized for a different control range.
- Use ***straight pipe runs*** of least 6 pipe diameters between control valve and fittings both upstream and downstream. Design for minimum pressure drop at fittings. Avoid sudden expansions and contractions in general. At pipe junctions, wyes and tees, use gradual (large radius) transitions wherever possible. Replace tees with wyes whenever possible. In general, the fittings with lowest pressure drop will produce the least noise.
- ***Select the smallest diameter valve*** that will carry and control the maximum flow expected.
- ***Avoid anomalous flow conditions***: avoid operating a valve at less than 30% of its rated capacity.
- ***Use valves with high values of F_L*** near full capacity. A given valve typically has better pressure recovery performance near full capacity than at minimum capacity.
- ***Special low-noise valves*** incorporate high values of F_L and, in some cases, built-in valve trim. A noise reduction benefit of 15 to 25 dB(A) is achievable with proper selection.
- ***Install an in-line silencer:*** The effectiveness of an in-line silencer is estimated at 20 dB(A). This applies to both noise within the pipe and noise radiated from the pipe up- or downstream of the silencer. Typical practice is to place the silencer downstream of the valve. Experience has shown that a downstream silencer alone brings a benefit of only about 10 dB(A) in some cases because the sound upstream of the valve remains unattenuated. In order to realize the full 20 dB(A) benefit of the silencer, both upstream and downstream silencers may be necessary. The piping between the valve and silencer should be selected with thick walls and perhaps be covered with lagging.
- ***Increase wall thickness of pipe***. Doubling the pipe wall thickness could bring a 5 dB(A) reduction.
- ***Coordinate f_p and pipe TL*:** Select valve diameter, pipe diameter and thickness so that the peak frequency f_p is several times greater than f_0, and preferably greater than f_r. Failing this, f_p should be less than f_c. Avoid selecting f_p similar to f_c.

- ***Use pressure reducing plates*** or valve trim to create smaller jets, thereby increasing peak frequency f_p relative to f_r. Smaller jet diameters usually take better advantage of pipe transmission loss by increasing f_p away from f_c[iii].

5.2.2. Measurement Orifices and Venturis

- ***Reduce pressure drop***: the measurement orifice or venturi with lowest pressure drop is desired. This means maximizing the diameter ratio A_O/A_i and using gradual inlet and discharge angles for venturis.
- ***Use straight pipe runs*** of least 6 pipe diameters between control valve and fittings both upstream and downstream. Note that reducing large-scale turbulence in this manner is also important for measurement accuracy.
- ***Apply lagging*** to the exterior of the pipe. Focus on the area downstream of the valve.

5.2.3. Vacuum Vents

- A series of ***pressure-reducing plates*** may be considered.
- ***Use a well-rounded inlet.*** Avoid obstructions or sharp edges in or near the throat.
- In vacuum blowdown applications, ***lengthen the blowdown time*** by reducing the mass flow rate.
- ***Apply lagging*** to the exterior of the pipe.

5.3. Structural Fatigue Criterion

High sound levels and the accompanying vibration make structural fatigue of valve parts a possibility. Valve manufacturers recommend that valve noise at 1 meter from the pipe wall be limited to 115 to 120 dB(A) to avoid fatigue. Note that in-line silencers or lagging are not helpful at reducing vibration levels within the valve where the danger of fatigue is greatest. A more detailed discussion of structural fatigue is presented in Section 8.1.3, page 8-2.

The Control Valve spreadsheet calculates a structural fatigue criterion based on sound power level within the pipe and compares it to computed in-pipe conditions. If interior sound levels are within 10 dB of Structural Fatigue Criterion, design alternatives that reduce noise at the source should be considered.

[iii] In cases where f_c is less than f_p, the addition of valve trim may be detrimental.

5.4. Noise Emission Estimation Using Workbook

Spreadsheets:

- Control Valves
- Orifices, Venturis and Vacuum Vents

Required Inputs:

- General: Gas Compressibility Factor Z, mass flow rate m'
- Upstream conditions: pressure P_1, temperature T_1,
- Valve, Piping and Nozzle: valve coefficient C_V, valve diameter D_v, downstream pipe diameter D_D, upstream pipe diameter D_U, pipe wall thickness t_p, orifice or venturi outer diameter D_o, orifice or venturi inner diameter D_i
- Downstream conditions: pressure P_2, temperature T_2,
- Observer: distance r, angle θ

Notes:

- The Spreadsheet performs a rudimentary valve sizing algorithm for gases. Select valve type using the scrolling box in Line 2a (note that the same type may be listed several times for various service conditions). The C_V and D_V of the valve is estimated. The user must enter the actual C_V selected. Consult valve manufacturers for greater accuracy in sizing.
- Sound power levels internal to the pipe are compared to the structural fatigue limits for the given pipe diameter in Part 4.
- The user may elect the inclusion of various control-valve related noise control elements, including valve trim, in-line silencers upstream and/or downstream of the valve, and downstream resistance plates. Note that to use the in-line silencer selection here it is not necessary to refer to the Silencers spreadsheet (See Section 8.4). The silencer performance used here is generic.

5.5. Noise Emission Equations for Control Valve Noise

The predictive equations for noise emission below follow the approach of Baumann[18] as adapted by Bies and Hansen[19] and Beranek and Ver[17]. A similar approach is adopted in various standards[20,21]. The user should be aware that most valve manufacturers incorporate noise prediction into their sizing software.

The overall sound pressure level inside the pipe is estimated as:

$$L_{pi,overall} = -56 + 10\log\left(\frac{\eta C_v F_L P_1 P_2 c_0^{\ 4} G^2}{D_v^{\ 2}}\right)$$

$$F_L^{\ 2} = \frac{P_1 - P_2}{P_1 - P_0}$$

where C_V is in customary units (gals/min per psia^{½}), η, F_L and G are dimensionless, P_1, P_2 and P_0 (the pressure in the vena contracta) are in newtons/meter^2 and D_v is in meters. The ratio $C_V/D_V^{\ 2}$ (where D_V is in millimeters), pressure recovery coefficient F_L, and valve style modifier F_D are tabulated below in Table 5 (page 5-7). The efficiency of conversion η depends on the stream Mach number M_j, as shown below in Figure 4.

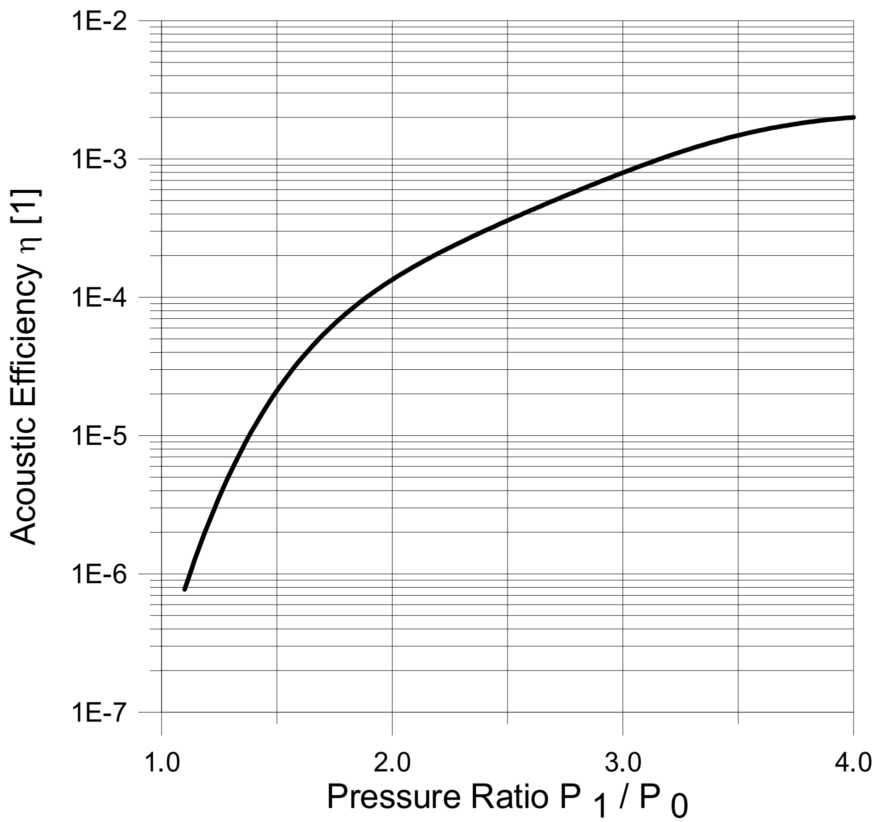


Figure 4: Acoustic Conversion Efficiency vs. Mach Number

Table 5: Typical Constants Associated with Control Valves

Type	Flow To	% Travel	C_V/D_V^2*	F_L	F_D
Globe, single-port parabolic plug	Open	100%	0.020	0.90	0.46
Globe. Single-port parabolic plug	Open	75%	0.015	0.90	0.36
Globe. Single-port parabolic plug	Open	50%	0.010	0.90	0.28
Globe. Single-port parabolic plug	Open	25%	0.005	0.90	0.16
Globe. Single-port parabolic plug	Open	10%	0.002	0.90	0.10
Globe. Single-port parabolic plug	Close	100%	0.025	0.80	1.00
Globe, V-port plug	Open	100%	0.016	0.92	0.50
Globe, V-port plug	Open	50%	0.008	0.95	0.42
Globe, V-port plug	Open	30%	0.005	0.95	0.41
Globe, four-port cage	Open	100%	0.025	0.90	0.43
Globe, four-port cage	Open	50%	0.013	0.90	0.36
Globe, six-port cage	Open	100%	0.025	0.90	0.32
Globe, six-port cage	Open	50%	0.013	0.90	0.25
Butterfly valve, swing-through vane	N/A	75º open	0.050	0.56	0.57
Butterfly valve, swing-through vane	N/A	60º open	0.030	0.67	0.50
Butterfly valve, swing-through vane	N/A	50º open	0.016	0.74	0.42
Butterfly valve, swing-through vane	N/A	40º open	0.010	0.78	0.34
Butterfly valve, swing-through vane	N/A	30º open	0.005	0.80	0.26
Butterfly valve, fluted vane	N/A	75º open	0.040	0.70	0.30
Butterfly valve, fluted vane	N/A	50º open	0.013	0.76	0.19
Butterfly valve, fluted vane	N/A	30º open	0.007	0.82	0.08
Eccentric rotary plug valve	Open	50º open	0.020	0.85	0.42
Eccentric rotary plug valve	Open	30º open	0.013	0.91	0.30
Eccentric rotary plug valve	Close	50º open	0.021	0.68	0.45
Eccentric rotary plug valve	Close	30º open	0.013	0.88	0.30
Ball valve, segmented	Open	60º open	0.018	0.66	0.75
Ball valve, segmented	Open	30º open	0.005	0.82	0.63

The peak frequency f_p of the noise spectrum depends on the velocity of the flow and the diameter of the jet D_j as

$$f_p = \frac{0.2 M_j c_0}{D_j} \quad M_j < \sqrt{2}$$

$$f_p = \frac{0.28 c_0}{D_j \sqrt{M_j^2 - 1}} \quad M_j \geq \sqrt{2}$$

where M_j is the Mach Number of the flow in the jet, D_j is the diameter of the jet (not the valve body or the pipe), and c_0 is the sonic velocity in the vena contracta.

The jet diameter may be estimated as

$$D_j \approx 4.6 \times 10^{-3} F_d \sqrt{C_V F_L}$$

where F_d is termed the "valve style modifier", empirically determined, and tabulated above in Table 5.

The stream Mach Number M_j is calculated as follows:

$$M_j = \left[\frac{2}{\gamma - 1} \left(\left(\phi \frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \right]^{\frac{1}{2}}$$

where

$$\phi = \left(\frac{\gamma + 1}{2} \right)^{\frac{\gamma}{\gamma-1}} - F_L^2 \left(\left(\frac{\gamma + 1}{2} \right)^{\frac{\gamma}{\gamma-1}} - 1 \right)$$

The level L_{pi} of the one-third octave band containing the spectrum peak frequency is $L_p = L_{pi,overall} - 8$. For frequencies greater than the peak frequency the spectrum rolls off at the rate of 3.5 dB per octave. For frequencies less than the peak frequency the spectrum rolls off at the rate of 5 dB per octave for the first two octaves and then at the rate of 3 dB per octave at lower frequencies.

The one-third octave band sound pressure levels external to the pipe at one meter from the pipe centerline are calculated as:

$$L_p \big|_{1m} = L_{pi} - TL + 5 + L_g$$

where

$$L_g = -16 \log \left(1 - \frac{1.3 \times 10^{-5} P_1 C_V F_L}{D_i^2 P_2} \right)$$

and TL is the pipe wall transmission loss. Circular pipe is generally quite effective at blocking the transmission of sound energy at both high and low frequencies. The weakest Transmission Loss occurs at an intermediate frequency, the first mode cut-on frequency of the pipe:

$$f_c = \frac{0.586 c_2}{D_p}$$

where c_2 is the sonic velocity downstream of the valve and D_p is the internal diameter of the pipe.

Best results for noise emission through the pipe wall are obtained when the peak noise emission frequency f_p and the first mode frequency f_c are widely spaced. A more thorough discussion of pipe wall transmission loss (TL) is given below in Section 8.1.

5.5.1. Noise Emission Equations for Measurement Orifices

From a noise emission standpoint, a measurement orifice can be treated as a special case of a control valve. It is essentially a high recovery valve with fixed control position. The values of C_V and F_L obtained below may be substituted into the control valve noise emission equations above.

The value of C_V is estimated from a general relationship with K:

$$C_V = \frac{D[mm]^2}{21.7\sqrt{K}}$$

$$K = \frac{\Delta P}{\frac{1}{2}\rho u^2}$$

K values for sudden contraction and expansion[22] can be estimated as

$$K = 1.53 - 2.58 \, {}^{A_i}\!/_{A_o} + 1.08 \left({}^{A_i}\!/_{A_o} \right)^2 .$$

where D is the diameter in millimeters at the flow constriction.

Applying isentropic expansion relations and a polynomial curve fit, it can be shown that F_L takes on the following approximate values:

$$F_L = 0.19 + 1.22 \frac{A_i}{A_o} - 0.612 \left(\frac{A_i}{A_o} \right)^2$$

5.5.2. Venturis

From a noise emission standpoint, a venturi is also similar to a control valve.

For 20° expansions and contractions[3], K can be estimated as

$$K = .82 - 1.18\, {A_i}/{A_o} + .352\left({A_i}/{A_o}\right)^2$$

The C_V value is therefore approximately

$$C_V \approx \frac{D[mm]^2}{21.7\sqrt{.82 - 1.18\, {A_i}/{A_o} + .352\left({A_i}/{A_o}\right)^2}}$$

where D is the diameter in millimeters at the flow constriction. From isentropic expansion relations it can be shown that an approximate F_L value is

$$F_L = 1.45\sqrt{1 - \left(\frac{\gamma - 1}{2}\frac{A_i}{A_o} + 1\right)^{-\gamma/\gamma - 1}}$$

when Tap 2 is downstream, or F_L = 1.000 when Tap 2 is at the vena contracta.

5.5.3. Intake (Vacuum) Vent

A high-velocity intake vent opening is also treated as a special case of a control valve[iv]. Gas accelerates into the opening and in many cases reaches sonic velocity. The resulting jet and possible shock waves are constrained within the pipe. The vacuum vent is treated as a low recovery valve with fixed control position. Based on K values for various inlet geometries[22], the effective C_V and F_L have been estimated and are tabulated below in Table 6.

[iv] Note that this analysis refers to the opening itself and not to control valves governing the flow downstream.

Table 6: C_V **and** F_L **factors for Intake Vent by Geometry**

Inlet Geometry	C_V	F_L
Well-rounded	$0.20 \times D[mm^2]$	1.0
Slightly-rounded	$0.09 \times D[mm^2]$	0.9
Projecting Pipe, L/D < 0.5	$0.06 \times D[mm^2]$	0.8
Projecting Pipe, L/D = 0.5	$0.05 \times D[mm^2]$	0.7
Projecting Pipe, L/D > 0.5	$0.04 \times D[mm^2]$	0.6

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6. GAS MOVING EQUIPMENT AND FLOW-STRUCTURE INTERACTIONS

Flow-structure interactions covered by this guide include Air Inlet Debris Screen, Compressors and Exhausters, Fans and Blowers, and Flow Noise from Pipes and Fittings.

Large-scale vortices in undisturbed flow produce aerodynamic quadrupoles and noise having a predominantly low frequency character without a definable peak. In the absence of obstacles in the flow, this type of noise dominates. Additional noise is generated through turbulence whenever flow encounters a structure. Dipole sources of differing characters are created and may become the dominant noise generation mechanism.

6.1. Noise Generation by Inlet Debris Screen and Fixed Obstructions

Narrow-band tonal sound is generated by vortices in the oscillating wake of slender obstructions such as wires, pipes, and struts. Although each individual vortex produces only shear forces, the succession of vortices with alternating rotational sense (Karmann vortex street) produces a series of dipoles that radiate with peak frequency

$$f_p = 0.2\frac{U}{D},$$

where U is the characteristic velocity of the flow and D is the characteristic dimension of the obstruction. Significant levels of upstream turbulence can increase noise emission.

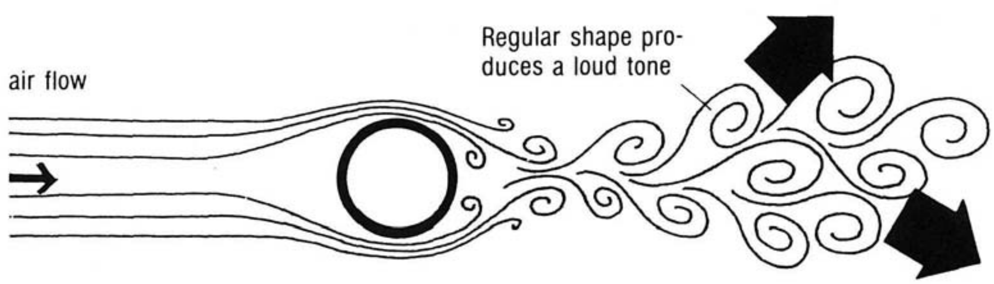


Figure 5: Vortex Street

(Ingemansson and Folkesson[23])

6.2. Noise Generation by Rotor-Stator Interactions

Lifting surfaces, such as those employed in compressors, exhausters, fans and blowers also creates dipoles. When the dipoles rotate with machinery, the repetitive pattern of fluctuating pressures passing a given point produces a narrow band tone known as the *blade passage tone*. This tone and its integer harmonics are a hallmark of this type of equipment. The intensity of this tone increases strongly with total static pressure rise across the blades.

The rotational speed of the pressure patterns from the rotor-stator interactions is

$$N_q = \frac{nBN}{q}$$

where N_q is the rotation rate of the q-th rotating pressure pattern
N is the rotation rate of the rotor shaft
B is the number of rotor blades
V is the number of stator blades
n and k are positive integer numbers
and $q = nB \pm kV$

For small values of the denominator (i.e., when kV is subtracted), the tangential speed of the pressure pattern at the *ND/2* exceeds the blade tip speed, and can approach sonic velocity. Noise emission increases dramatically under these conditions. This can be avoided by designing so that q is large.

The frequency of the tone emitted by the q-th pressure pattern is simply the n-th harmonic tone:

$$f_q = nBN$$

6.3. Noise Generation by Compressors and Exhausters

Quadrupole radiation from large-scale vortices in the flow produces a broadband noise spectrum, to which are added a tone or tones related to mechanical action.

In reciprocating equipment a single low frequency tone is produced that corresponds to the number of pressure pulses produced per second. A rotary lobe compressor also produces essentially one mid-frequency tone, but the frequency is somewhat higher.

For rotating machinery such as axial and centrifugal compressors, the tones are typically high frequency and are produced by blade passage and rotor-stator interactions. The sound power developed depends on blade tip speed to the fifth power and horsepower squared.

6.4. Noise Generated by Fans and Blowers

The mechanism of fan noise generation is similar to that for compressors and exhausters, the main difference between the equipment being the pressures developed. Because fans and blowers are typically low-pressure devices, their mechanical and acoustical performance is strongly influenced by downstream conditions.

Broadband noise generation is a function of the blade type, flow, total static pressure rise, and operating point. Fan scaling laws relate the flow, pressure developed and acoustic power output to the rotation rate and diameter of homologous fans at the same operating point as follows:

Table 7: Fan Scaling Laws

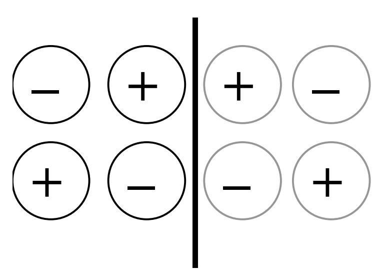
Blade Tip Speed = ND
Flow $\propto ND^3$
Total Static Pressure Rise $\propto N^2D^2$
Power Transmitted to Flow $\propto N^3D^5$
Acoustic Power $\propto$ Flow $\times$ Pressure^2 $\propto N^5D^7$

The fan scaling laws show that the ratio of acoustic power to mechanical power is proportional to the total static pressure rise. From this it follows that minimizing system pressure losses can help reduce noise emission. With system pressure reduced, it is usually possible to select a larger fan rotating more slowly to deliver the required flow. This is especially true for the blade passage tone, which intensifies dramatically as the total static pressure rise increases.

Furthermore, it is possible to deduce a general rule regarding noise emission from fans. The tradeoff between diameter D and rotation rate N is very important. A larger fan turning more slowly is generally preferred, as long as it operates near maximum static efficiency.

Maximum static efficiency corresponds to maximum air movement for minimum mechanical work, and as expected corresponds to minimum specific noise emission (noise emission per work done) for a given fan.

6.5. Flow Noise in Pipes and at Fittings



Broadband noise is generated in the boundary layer clinging to pipe walls. Pressure fluctuations from large-scale vortices in the turbulent flow reflect from pipe walls, producing a reinforcing pair of oscillatory forces rather than an opposing pair. The result is a series of dipoles at the pipe perimeter. The spectrum of sound within the pipe is dominated by low frequencies and contains no peaks.

The roughness of the pipe and presence of fittings such as wyes and tees increases flow noise output. These elements can dramatically increase the turbulence level in the pipe and hence the mechancial power available to be converted into acoustic energy.

6.5.1. Casing-Radiated Noise

Casing-radiated Noise arises from flow-induced and acoustically-induced vibrations of the casing. For fans and blowers, the casing is often a thin piece of flat sheet metal. For compressors and exhausters, the casing is constructed primarily of thick curved plates.

Casing-radiated noise is usually not an issue unless the inlet and outlet openings and ductwork are effectively silenced.

6.6. Reduced-Noise Design for Inlet Debris Screen and All Fixed Obstructions

- ***Streamline objects in the flow path*.** Sharp edges in the flow should be avoided, because they can produce locally high flow velocities and shock waves that increase noise emission. A steam train whistle is a relevant example. Rounded leading edges and streamlined trailing edges should be employed on all flow obstructions. Structural supports should be devoid of projections such as screws, welds, etc.
- ***Trailing edge boundary layer trips*** may also be useful in destabilizing the vortex street.

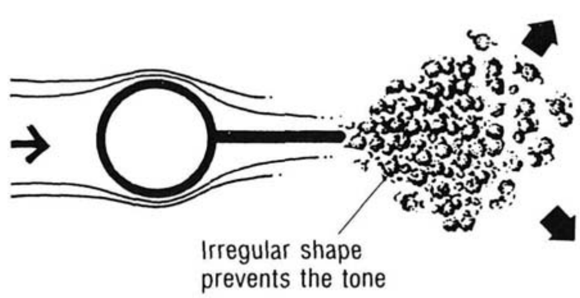


Figure 6: Reducing Vortex Tone

(Ingemansson and Folkesson[23])

- ***Minimize turbulence.*** Reducing turbulence minimizes the mechanical energy available for conversion into sound.
- ***Detune f_p from duct modes.*** The vortex shedding frequency of any flow obstacle should be selected below the first mode cut-on frequency of the pipe or duct. Above this frequency, resonant coupling between the vortex street and the sound field could lead to strong tones. The first mode cut-on frequency for a circular pipe is $0.586\ c/D$, and for a rectangular duct $0.500\ c/D_1$, where D_1 is larger of the two pipe cross-sectional dimensions.

6.7. Reduced-Noise Design for Gas-Moving Equipment

Useful references for further investigation in this area include Universal[24], NASA[25], and Burgess-Manning[26].

- ***Reduce turbulence***: allow at least one diameter of straight duct flow before a compressor or exhauster inlet (See Figure 7 below).
- ***Select a larger machine*** operating at lower RPM. This will probably require that system pressure losses be minimized.
- ***Design for a high lobe number q***. The number of rotors and stators, *B* and *V* respectively, should not be equal. Nor should they be related by near integers (e.g., 3 and 4). Larger prime numbers are preferred where *V* and *B* are widely spaced.
- ***Design for cutoff***: Choose *V* and *B* to achieve a cutoff factor less than 1.05. Fundamental tone is attenuated 8 dB. See Section 7.1 (page 7-1).

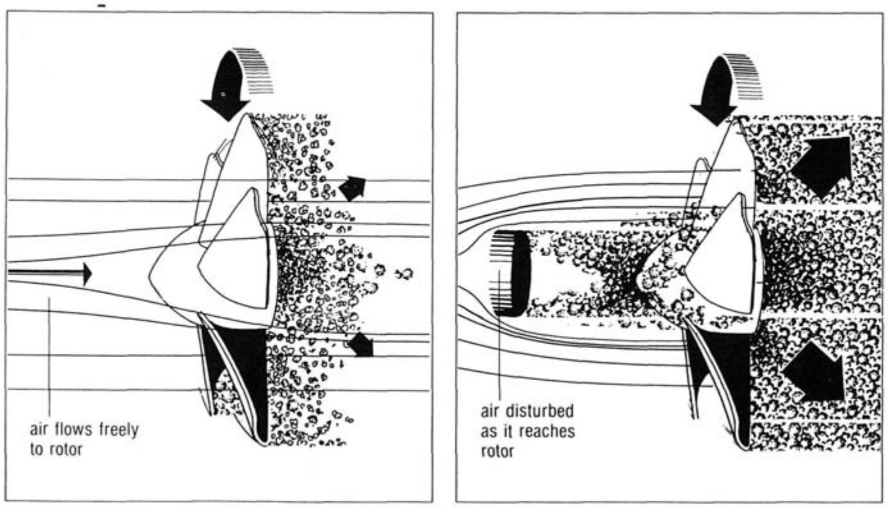


Figure 7: Effect of Turbulence Upstream of Rotor

(Ingemansson and Folkesson [23])

- ***Detune f_n from duct modes***: Select duct and impeller so that as many rotor/stator tones f_n, and at least the fundamental tone, lie below the first duct mode cut-on frequency. The first mode cut-on frequency for a circular pipe is $0.586\ c/D$, and for a rectangular duct $0.500\ c/D_1$, where D_1 is larger of the two pipe cross-sectional dimensions.
- ***Apply silencers***: Reciprocating and rotary lobe blowers are usually best serviced by reactive silencers because the silencers are more compact and do not require acoustical fill which could be degraded by oil mist in the discharge. Typical silencers for axial and centrifugal compressors are typically dissipative. Any exposed piping and ductwork between the unit and a silencer should be lagged.
- ***Modify casing***: once the inlet and discharge have been effectively silenced, casing noise may require attention. Additional stiffening members welded directly to the casing performs most effectively on flat plates and in general attenuates mainly low frequencies. Adding damping directly to the casing chiefly attenuates high frequencies if the thickness of the damping compound is comparable to the thickness of the material and if resonant radiation is present. Mass should only be added directly to the casing when the mass per unit area can be increased by at least 50%. In each case, an approximate 5 dB benefit is available in the associated frequency ranges.
- ***Apply acoustical lagging to casing and ducts***: Acoustical lagging consisting of a layer of sound insulation material (2-in., 4-in. or 6-in. thickness) and a limp, massive covering (1 psf) may be applied to the exterior of the casing, piping and

ductwork. Useful attenuation is available for high frequencies (1000 Hz and greater) for all thickness. Lower frequencies require thicker lagging.

- ***Vibration isolation*** may be necessary, in particular for reciprocating compressors and exhausters. Remember that vibrational energy can be converted into sound most efficiently by structures that are relatively thin and have large areas. Large equipment should therefore be sited on grade with a properly designed foundation block.
- ***Maximize Rotor/Stator spacing:*** Rotor/stator spacing should be at least 1.5 rotor chord widths. Further reductions of rotor/stator interaction tones, on the order of $2 \times RSS/C_2$ *dB(A),* can be achieved by further increasing spacing, where *RSS* is the rotor/stator spacing and C_2 is the rotor chord.

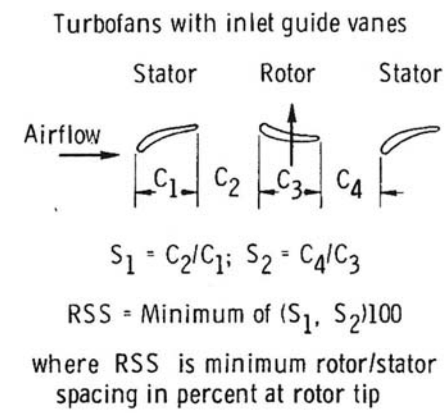


Figure 8: Rotor-Stator Spacing Coefficient

- ***Position stators downstream*** of the rotor whenever possible. Rotor-stator interactions are stronger for upstream stators than for downstream stators. Use inlet guide vanes only when required.

6.8. Reduced-Noise Design for Fans and Blowers

- ***Minimize system pressure losses.***
- ***Select a large fan rotating slowly.***
- ***Select the quietest wheel type*** appropriate for the service.
- ***Set the operating point*** within 5% of maximum static efficiency. A 1 dB(A) noise increase occurs for every 5% of max operating efficiency below max operating efficiency. [27]

- ***Use variable speed motors*** to control flow rather than inlet guide vanes, control valves, or other restrictive flow devices.
- ***Minimize upstream turbulence:*** Poor inflow conditions can lead to a condition called rotating stall, which produces a rumble and tone centered on 2/3 the shaft rotation frequency. Require 1.5 to 2.0 diameters of straight duct upstream of inlet.
- ***Avoid unstable flow regimes:*** Centrifugal and vaneaxial fans are unstable at operating points to the left of maximum static efficiency. In this region there are some pressures for which two different flow rates are possible. Operating in this region could cause the fan to surge back and forth between the two operating points. The surge frequency depends on the length of the attached piping.
- ***Select vibration isolation*** based on lowest rotational speed for variable speed systems.
- ***Orient discharge and downstream turns*** to have the same rotational sense as the flow. Otherwise, turbulence-induced rumble can result. If an elbow must be placed within 1.5 duct diameters of the discharge, the elbow shall have a long radius and incorporate turning vanes.

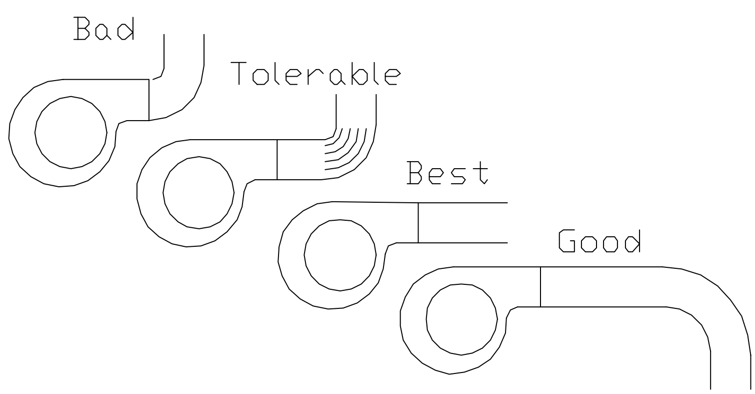


Figure 9: Proper Orientation of Discharge Turns

(after Schaffer[28])

6.9. Noise Reduction Recommendations for Flow

- ***Reduce flow velocity*** to approximately $85/\rho^{1/2}$ feet per second, where ρ is in pounds per cubic feet. "Economic velocity" for flow in the pipe may be somewhat lower. Consult relevant piping codes for other velocity limitations.

- ***Reduce number, abruptness and density of fittings*:** Prefer gradual transitions, welded over screwed or mitered bends, etc.
- ***Increase pipe wall thickness.*** Doubling the pipe wall thickness affords approximately 3 to 5 dB(A) additional attenuation.
- ***Apply acoustical lagging*** to the piping.

6.10. Noise Emission Estimation Using Workbook

Spreadsheets, with Required Inputs and Notes:

- Inlet Debris Screen

Required Inputs:

- A_S, m', D_W, POA, r

Notes:

- Percentage Open Area is that of the screen area not occluded by wires or other obstructions.

- Compressors and Exhausters

Required Inputs:

- W_M, D_I, N, B, L, H, W, r

Notes:

- For some equipment types casing noise and the noise from an unmuffled inlet are broken out separately. For others, they are reported together.

- Fans and Blowers

Required Inputs:

- Blower Type, m', P_{TS}, N, B, SE, PSE, Silencer IL, Silencer $L_{W,SN}$, L, H, W, r

Notes:

See Table 9 for K_W values for specific blower wheel types. If problems are occurring in a particular frequency band, look for the wheel type with the lowest K_W in that band.

- Flow Noise

Required Inputs:
• m', P_1, T_1, D_p, t_p, L, K, r
Notes:
• In Part 4, enter the number of each type of fitting that appears in a 10-ft. section of pipe. Technically, this refers to an individual 10-ft. long section. A coarse approximation for the overall system may be obtained by entering the average number of each fitting appearing per 10-ft. of pipe.

6.11. Predictive Equations for Inlet Debris Screen

The sound power level of noise emission from an inlet debris screen is estimated (after Beranek and Ver[29]) as

$$L_W = L_{W,overall} + F\left({f}/{f_p} \right)$$
$$L_{W,overall} = 10 + 10\log_{10}\left(S\xi^3 U^6\right) + 10\log_{10}\left(1 - M\right)$$

where S is the screen area in square meters, U is the flow velocity in meters per second, M is the Mach number of the flow, and ξ is an effective head loss coefficient combining the coefficient of drag of a small cylinder and the percent open area (POA) of the screen, equal to

$$\xi = 1.1\left(1 - \frac{POA}{100}\right).$$

The spectral shape function $F(f/f_p)$ is approximated by

$$F\left({f}/{f_p} \right) = -0.2 + .384\log_{10}\left({f}/{f_p} \right) - 1.09738\left(\log_{10}\left({f}/{f_p} \right)\right)^2$$

6.12. Predictive Equations for Compressors and Exhausters

Noise emission equations for compressors and exhausters are taken from Heitner[30] as given in Bies and Hansen[31]. The equations are believe to be equally valid for use with exhausters.

6.12.1. Centrifugal Compressors

For centrifugal compressors and exhausters, the overall sound power level measured at the discharge piping inside the pipe is given by[31]

$$L_{W,overall} = 20\log_{10} W_M + 50\log_{10} U - 45,$$
$$L_W = L_{W,overall} + F(f)$$

where U is the impeller tip speed in meters per second (limited to the range 30 to 230 ms^{-1}) and W_M is the mechanical power of the drive motor in kilowatts. The peak frequency is[31]

$$f_p = 4.1U \quad [Hz].$$

The spectrum level in the octave band containing f_p is taken as 4.5 dB less than $L_{W,overall}$. The spectrum rolls off at the rate of 3 dB per octave above and below the peak frequency.

Noise estimates for the casing and for the unmuffled inlet are[31]

$$L_{W,overall}\Big|_{\text{Casing}} = 79 + 10\log_{10} W_M$$
$$L_{W,overall}\Big|_{Inlet} = 80 + 10\log_{10} W_M$$

The spectral corrections of Table 8 are subtracted from the corresponding overall sound power level value to give octave band sound power levels.

Table 8: Octave Band Corrections for Compressor and Exhauster Inlets and Casings

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*
Centrifugal Casing	10	10	11	13	13	11	7	8	12
Centrifugal Inlet	18	16	14	10	8	6	5	10	16
Rotary and Recip. Inlet and Casing	11	15	10	11	13	10	5	8	15

6.12.2. Rotary or Axial Compressors

The overall sound power level at the pipe exit may be estimated as[31]

$$L_{W,overall} = 68.5 + 20\log_{10} W_M$$

The peak frequency is that of the second blade harmonic

$$f_p = 2BN$$

where N is the number of rotations per second and B the number of blades.

The sound power spectrum is assembled from estimates for the 63 Hz and 500 Hz octave bands, the octave band containing f_p and the octave band containing the frequency $f_h = f_p^2/400$: [31]

$$L_W\big|_{63} = 76.5 + 10\log_{10} W_M$$
$$L_W\big|_{500} = 72 + 13.5\log_{10} W_M$$
$$L_W\big|_{p} = 66.5 + 20\log_{10} W_M$$
$$L_W\big|_{h} = 72 + 13.5\log_{10} W_M$$

A straight line is drawn between these points and the slope is continued for octave bands outside these points.

Casing noise (including partially muffled air inlets) is estimated as[31]:

$$L_{W,overall} = 90 + 10\log_{10} W_M$$

The frequency corrections given in Table 8 are subtracted from overall sound power levels to give the octave band sound power level values.

6.12.3. Reciprocating Compressors

The overall sound power level in the exit piping of a compressor can be estimated as:

$$L_{W,overall} = 106.5 + 10\log_{10} W_M$$

The peak frequency is that of the cylinder frequency

$$f_p = BN$$

where B is here the number of cylinders. The spectrum level in the octave band containing f_p is taken as 4.5 dB less than $L_{W,overall}$. The spectrum rolls off at the rate of 3 dB per octave above and below the peak frequency.

Casing noise (including partially muffled air inlets) is as given above under Rotary Compressors.

6.13. Noise Emission From Fans and Blowers

Fan and Blower noise is estimated according to a method developed initially by Buffalo Forge[27] and published later in ASHRAE[32]. Bies and Hansen[31] added a correction for static efficiency. The octave band sound power level of a fan or blower is estimated as

$$L_W = K_W + 10\log_{10} Q + 20\log_{10} P_{TS} + \left(\frac{.95 - \frac{SE}{PSE}}{.05} \right)$$

where Q is flow rate in cubic feet per minute, P_{TS} is total static pressure rise across the fan in inches of water column, and K_W is tabulated in Table 9 for various fan types.

The column *BFI* is the Blade Frequency Index, which is an increment added to the octave band containing the blade passage frequency,

$$f_b = BN .$$

Table 9: Specific Sound Power Level K_W by Fan Type

Wheel Type	***31.5***	***63***	***125***	***250***	***500***	***1000***	***2000***	***4000***	***8000***	***BFI***
Centrifugal, AF, BC, or BI, D > 30"	37	37	37	36	31	27	20	16	14	3
Centrifugal, AF, BC, or BI, D < 30"	42	42	42	40	36	31	25	21	16	3
Centrifugal, FC, All Sizes	50	50	50	40	33	33	28	23	18	2
Radial, 4" - 10" SP, D > 40"	53	53	44	40	36	34	29	26	23	7
Radial, 4" - 10" SP, D < 40"	64	64	56	50	40	39	36	31	28	7
Radial, 10" – 20" SP, D > 40"	55	55	51	42	39	35	30	26	23	8
Radial, 10" – 20" SP, D < 40"	65	65	60	48	45	43	38	34	31	8
Radial, 20" – 60" SP, D > 40"	58	58	55	50	45	43	41	38	35	8
Radial, 20" – 60" SP, D < 40"	68	68	64	56	51	51	49	46	43	8
Vaneaxial, Hub Ratio 0.3 to 0.4	46	46	40	40	45	44	42	35	13	6
Vaneaxial, Hub Ratio 0.4 to 0.6	46	46	40	43	40	38	33	27	25	6
Vaneaxial, Hub Ratio 0.6 to 0.8	56	56	49	48	48	46	44	40	37	6
Tubeaxial, D > 40"	48	48	43	44	46	44	43	36	34	7
Tubeaxial, D < 40"	45	45	44	46	50	49	48	40	37	7
Propeller, D < 12 ft.	45	45	48	55	53	52	49	43	39	5

6.14. Noise Estimation for Flow in Pipes

The following method was suggested by Seebold (1973)[33] and continues to receive widespread acceptance. The method estimates the noise that results from the boundary layer pressure fluctuations in fully developed flow in uninterrupted straight circular pipes, and then applies a loss-factor correction *K* for local discontinuities. The Sound Pressure Level (presumably at 1 meter from the pipe) in the octave band centered on frequency f is estimated from the flow velocity, gas density ρ, pipe thickness *T* and diameter *D*, ring frequency f_r of the pipe, and a spectral correction *S:*

$$L_P\big|_{1m} = 40\log_{10} U + 20\log_{10}\rho + 20\log_{10} K - 10\log_{10}\left[\frac{t_p}{D_p}\left(1+\frac{6}{D_p}\right)\right]\ldots$$

$$\ldots - 5\log_{10}\left|\frac{f}{f_r}\left(1-\frac{f}{f_r}\right)\right| + \Delta L_P$$

where U is in feet per second, ρ is in pounds per cubic foot, t_p and D_p are in feet, and f and f_r are in Hertz. The ring frequency for steel pipe is approximately *5275/D*.

The spectral correction ΔL_P depends on ratio of the octave band center frequency f to the peak frequency f_p as

$$\Delta L_P = \begin{cases} 11.4\log_{10}\dfrac{f}{f_p} + 10.4 & \text{if } \dfrac{f}{f_p} < 0.5 \\ 7 & \text{if } 0.5 \le \dfrac{f}{f_p} < 5 \\ -10\log_{10}\dfrac{f}{f_p} + 14 & \text{if } 5 \le \dfrac{f}{f_p} < 12 \\ -36.1\log_{10}\dfrac{f}{f_p} + 41.9 & \text{if } \dfrac{f}{f_p} \ge 12 \end{cases}$$

The loss-factor *K* is determined by adding the individual loss factors K_i for the flow fittings and elements present within a 10 ft. length of pipe. The loss-factors K_i are tabulated below in Table 10.

The most correct way to perform this estimation is to evaluate each individual 10-foot segment. The aggregate noise emission is computed from the sum of the individual noise emissions (see Appendix B).

An alternative method is to compute *K* based on the average number of components appearing in a 10-foot section over the length of the piping (e.g., 0.8 ninety-degree turns per 10-foot section would be entered where 4 turns are present in a 50 foot piping run). The estimated noise emission should then be assumed to be present along the evaluated length. Note that the latter method does not allow identification of localized noise sources and hot spots.

Table 10: Loss Factors K_i for Pipe Flow Noise

Straight Pipe		0.12				
45° Elbow	Screwed	0.42	Welded, R/D=1	0.20	Welded, R/D=1.5	0.11
90° Elbow	Screwed	0.98	Welded, R/D=1	0.45	Welded, R/D=1.5	0.32
180° Elbow	Screwed	3.00	Welded, R/D=1	0.80	Welded, R/D=1.5	0.43
Tees (Screwed)	Thru Branch	1.80	Thru Run	0.50		
Tees (Welded)	Thru Branch	1.40	Thru Run	0.40		
Reducer	D2/D1= 0.3	0.25	D2/D1= 0.5	0.17	D2/D1= 0.7	0.07
Expander	D2/D1= 3	0.8	D2/D1= 2	0.58	D2/D1= 1.25	0.1
Sudden Contraction	D2/D1= 0.1	0.48	D2/D1 = 0.33	0.41	D2/D1 = .80	0.12
Sudden Expansion	D2/D1= 10	0.98	D2/D1= 3	0.7	D2/D1= 1.25	0.12

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7. TURBOMACHINERY

The noise sources described in this section are related to the operation of turbomachinery. Aircraft engines and their derivatives are typically items of research interest at NASA Glenn Research Center. Although reducing noise of these components is an integral part of GRC's work, they are not candidates for the industrial noise control methods in this *Guide*. The noise they generate may however have significant noise control ramifications and must be accounted for in the system design.

The aircraft engine is considered as a combination of separate components. Components addressed include:

- Inlet Fan and Compressor
- Combustor and Core
- Turbine
- Jet Mixing
- Jet Shock-Associated Noise

Noise emission predictions are based on empirical correlation studies. Superior estimates may also be available within computer models developed by Clark[34]. Related methods are described in *Aeroacoustics of Flight Vehicles*[35].

7.1. Inlet Fan and Compressor

Inlet fan and compressor noise generation in an aircraft engine differs little from other industrial axial compressors, with the exception of the first stage high bypass ratio fan, the absence of a long inlet duct, and a different approach to expressing the design parameters.

The noise emission estimates given below could be used for in-duct sound power of large industrial axial compressors if one integrates the power over all angles. In this case select an observation angle of 0° to get an appropriate sound power level estimate.

Noise emission estimates are computed after a NASA Glenn model by Heidmann[36]. The noise emission is shown to be related to the work performed by the fan and compressor, as expressed by the temperature rise or pressure ratio, the mass flow rate, the tip Mach number M_{TR} and design tip Mach number M_{TRD}, rotor/stator spacing and distance and direction of observation.

Broadband and tonal noise is estimated for both inlet and discharge. Combination tones are estimated for first stage fans. In the Workbook, the assumption has been made that discrete tones are increased due to additional turbulence experienced in static test stand operations.

Because the noise control of these devices is outside the scope of the *Design Guide*, and because the source documents should be readily available at NASA Glenn Research Center, the rather lengthy equations have been omitted.

The basic noise emission equation for all broadband and tonal noise estimates is of the form:

$$L_p = 20\log_{10}\left(\frac{\Delta T}{{}^\circ R}\right) + 10\log_{10}\left(\frac{\dot{m}}{1\,\text{lbm/sec}}\right) + F_1\left(MTR, MTRD\right) + F_2\left(RSS\right) + F_3\left(\theta\right) + \ldots$$
$$\ldots + F_4\left(f/f_b\right) - 20\log_{10}\left(\frac{r}{1\,\text{m}}\right)$$

The inlet noise peaks at an angle about 30° from the inlet, and discharge noise about 110° from the inlet. The most useful area for noise reduction is represented by the F_2 term, where reductions of the order of 5 and 10 dB(A) are possible by increasing rotor/stator spacing. Another potential area for reducing noise involves arranging the rotor/stator interaction to achieve cutoff. A simplified approach to establishing the cutoff condition is

$$\delta = \left|\frac{M_{TR}}{1 - V/B}\right| \leq 1.05$$

where V is the number of stators (vanes) and B the number of rotors (blades). When the cutoff condition exists, the fundamental blade passage tone is reduced 8 dB.

7.2. Combustor and Core noise

Combustor and Core noise is estimated using the method of ARP 876C (1985)[39].

The overall sound power level is estimated from the mass flow rate m', combustor inlet total pressure P_3, combustor total temperature rise $T_4 - T_3$, reference total temperature extraaction by the turbines at maximum takeoff conditions $(T_4\text{-}T_5)_{ref}$, and the temperature, pressure and sonic velocity for sea level standard conditions.

The noise emission varies as 10 $\log_{10} m'$ and as 20 $\log_{10} (T_4 - T_3)$ and 20 $\log_{10} (P_3)$. The peak frequency f_p is apparently always close to 400 Hz and the farfield radiation is only moderately directional, peaking at an angle of 60° from the inlet.

Because the noise control of these devices falls outside the scope of the *Design Guide*, the equations have been omitted.

7.3. Turbine Noise

Turbine noise is estimated following the recommendations of Krejsa and Valerino[37]. The sound pressure level at a radius of 47.5 meters (150 ft.) is estimated using the relative tip speed of the last rotor V_{TR}, the sonic velocity at the exit c_L, the primary mass flow m. Estimates are provided for both broadband and tonal content. The noise spectrum peaks at the blade passage frequency f_b and at an angle of 110° from the inlet. The implementation of these equations in the Workbook assumes that the primary nozzle exit is not upstream of the secondary nozzle exit. Reductions of up to 10 dB(A) can be achieved if the primary nozzle exit is located upstream of the secondary nozzle as in a JT8D engine.

The tone SPL varies with $10 \log_{10} C/S$ where C and S are respectively the stator chord length and rotor/stator spacing at the final rotor. A 3 dB reduction is available by doubling the spacing (halving the ratio C/S).

Because the noise control of these devices falls outside the scope of the *Design Guide*, the equations have been omitted.

7.4. Jet Noise

Three principal noise source mechanisms exist: mixing, shock-associated noise, and screech. Noise esimates are based on the method of Stone and Montagni[40] as reported by SAE ARP 876C[39] and Beranek and Vér[38].

Mixing noise arises at the turbulent shear layer separating the fast moving jet core from the stationary surrounding atmosphere. Shock-associated noise arises in choked flows, and dominates above $M_j = 1$. A third source is jet "screech", produced by a feedback mechanism in which a disturbance convected in the shear layer generates sound as it traverses the standing system of shock waves. The sound propagates upstream through the ambient atmosphere and causes the release of a new flow disturbance at the nozzle exit. This is amplified as it convects downstream and the feedback loop is completed as it encounters the shocks.

7.5. Noise Estimation Using Workbook

Spreadsheets, Inputs Required and Notes:

- Inlet Fan and Compressor

> Inputs Required: mass flow rate m', upstream pressure P_1, upstream temperature T_1, downstream pressure P_2, diameter of fan D_F, rotational rate N, number of blades B, number of vanes or stators V, inlet guide vane chord length C_1, inlet guide vane/fan rotor spacing S_1, fan rotor chord length C_2, rotor/stator spacing S_2, M_{TRD}, Fan Stage, distance r, angle θ
>
> Notes:
>
> - Levels of broadband and tonal noise are tabulated separately for radiation to the observation point from inlet and outlet. Levels of combination tones are tabulated for the inlet.

- Combustor and Core

Inputs Required: mass flow rate m', combustor inlet pressure P_3, combustor inlet temperature T_3, combustor outlet temperature T_4, reference turbine temperature differential for maximum takeoff conditions $(T_4\text{-}T_5)_{ref}$, distance r, angle θ

- Turbine Noise

Inputs Required: mass flow rate m', turbine exit temperature T_5, turbine diameter D_T, rotational rate N, number of blades B, rotor chord length C, rotor/stator spacing S, distance r, angle θ

Notes:

- Broadband and tonal noise are tabulated separately.
- Turbine noise is assumed to radiate exclusively from the engine discharge.

- Jet Mixing

Inputs Required:

- Upstream Gas Conditions: pressure P_1, temperature T_1
- Downstream Gas Conditions: pressure P_a, temperature T_a
- Nozzle: nozzle coefficient C_N, nozzle diameter D_N,
- Observer: distance r, angle θ

Notes:

- Gas is selectable so that this method may be used with all forms of gas discharge.
- The "Execute" button must be pushed (clicked-on) in order to perform the double summation function for Shock-Associated Noise when $M_j>1$. Failing to do this will cause Shock-Associated Noise to be left out of the computations.
- Jet Mixing Noise and Jet Shock Noise results are tabulated separately.
- Use a nozzle coefficient of 0.85 if C_N is not known.
- Do not expect the estimated noise levels to meet a hearing conservation criterion.

7.6. Noise Estimation for Jet Mixing Noise

It is customary to express the parameters of the gas flow as if it were an ideal, expanded jet with isentropic characteristics. The jet parameters for an ideal expanded jet can be calculated from the upstream and downstream pressures and temperatures:

$$M_j = \sqrt{\frac{2}{\gamma - 1}\left(\left(\frac{P_1}{P_2}\right)^{\frac{\gamma}{\gamma-1}} - 1\right)}$$

$$T_j = \frac{T_1}{1 + \frac{\gamma + 1}{2} M_j^{\,2}}$$

$$c_j = \sqrt{\gamma \frac{R}{MW} T_j}$$

$$u_j = M_j c_j \quad ; \quad M_c = 0.62 M_j$$

The overall sound pressure level of jet mixing noise measured at an angle of 90° from the jet axis can be estimated as[39,40,41].

$$L_{P,overall}(90^\circ) = 140 + 10\log\left(\frac{A_j}{r^2}\left(\frac{P_2}{P_{ISA}}\right)^2\left(\frac{\rho_j}{\rho_2}\right)^w\right) + 10\log\left(\frac{M_j^{\,7.5}}{1 - 0.1M_j^{\,2.5} + 0.015M_j^{\,4.5}}\right)$$

where A_j is the fully expanded jet area ($\pi/4\ D_N^2$ for a subsonic jet), r is the distance to the observation point, P_{ISA} refers to standard atmospheric pressure, and

$$w = \frac{3M_j^{\,3.5}}{0.6 + M_j^{\,3.5}} - 1$$

For other angles:

$$L_{P,overall}(\theta) = L_{P,overall}(90^\circ) - 30\log\left(1 - \frac{M_c \cos\theta}{\left(1 + M_c^{\,5}\right)^{\frac{1}{5}}}\right) - 1.67\log\left(1 + \frac{1}{10^{40.56-\theta'} + 4\times10^{-6}}\right)$$

$$\theta' = 0.26(180 - \theta)M_j^{\,0.1}$$

where θ is expressed in degrees relative to the discharge axis.

The peak frequency of the resulting noise spectrum is computed as

$$f_p = \frac{S_j U_j}{D_N}$$

where D_N is the nozzle exit diameter and S_j varies with T_j/T_a and θ, and is interpolated from Table 11.

The spectral shape is approximated by

$$\Delta L_P = -\Delta - 8.4\left(\log\left(\frac{f}{f_p}\right)\right)^2$$

where Δ is interpolated from Table 12 below. The values ΔL_P are added to $L_{P,overall}$ to give octave sound pressure level values.

Table 11: Values of Strouhal Number as a Function of T_j/T_a and θ

T_j/T_a	$\theta = 50°$	$\theta = 60°$	$\theta = 70°$	$\theta = 80°$	$\theta \geq 90°$
1	0.7	0.8	0.8	1.0	0.9
2	0.5	0.4	0.6	0.5	0.6
3	0.3	0.4	0.4	0.4	0.5

Table 12: Values of Δ as a Function of T_j/T_a and θ

T_j/T_a	$\theta = 50°$	$\theta = 60°$	$\theta \geq 70°$
1	11 dB	11 dB	11 dB
2	10 dB	10 dB	11 dB
3	9 dB	10 dB	10 dB

The above equations apply to all cold jets and to hot jets when observed from an angle more than 50° from the jet discharge axis.

For hot jets (T_j/T_a>1.1), the peak frequency and spectral shape are considerably altered for $\theta \leq 50°$. This is due to refraction of sound at the shear layer. Three simplified corrections to the spectral peak frequency are estimated from tables given in SAE ARP 876C[39]. A change in the spectral shape also occurs, but is considered less important for noise control purposes and is omitted here.

For hot jets the shift in peak frequency from the value calculated above is expressed as a number of ISO-preferred 1/3-octave bands ΔBN_i. (For cold jets no adjustments are necessary). The first ΔBN_1 depends on the angle of observation, the second ΔBN_2 on the ratio of jet temperature to ambient temperature and the third ΔBN_3 is an additional correction depending on angle that is used when $M_j > 1.33$.

$$\Delta BN_1 = -1.1 + 0.262\theta(\deg) - 1.543\times 10^{-4}\,\theta(\deg)^2$$

$$\Delta BN_2 = 1 - \frac{T_j}{T_s}$$

$$\textit{and if } M_j > 1.33,$$

$$\begin{aligned}\Delta BN_3 &= \frac{50-\theta}{10} \quad \textit{for} \quad 20^\circ < \theta \le 50^\circ \\ &= 3.0 \quad \textit{for} \quad \theta \le 20^\circ \\ &= 0.0 \quad \textit{for} \quad \theta > 50^\circ\end{aligned}$$

where θ is relative to the jet discharge axis. The shifted peak frequency f_p' for hot jets is

$$f_p' = f_p 10^{0.1\sum_i \Delta BN_i}$$

7.7. Predictive Equations for Shock-Associated Noise

Predictive equations for shock-associated noise follow the method reported in SAE ARP 876C[39] and Beranek and Ver[41]. Shock associated noise dominates for $M_j \ge 1$ in the absence of a converging-diverging nozzle, but is not generated at exit velocities $M_j < 1$. Shock-associated noise is essentially omni-directional and may be estimated for all angles of observation as follows:

$$L_{P\,shock} = C_0 + 10\log\frac{\beta^n A_j}{r^2}$$

where

$$\beta = \sqrt{M_j^{\ 2} - 1}$$

$$C_0 = 156.5 \quad for \frac{T_j}{T_s} < 1.1$$

$$and \quad n = 4 \quad for\ \beta < 1$$

$$n = 1 \quad for\ \beta \ge 1$$

$$C_0 = 158.5 \quad for \frac{T_j}{T_s} \ge 1.1$$

$$and \quad n = 4 \quad for\ \beta < 1$$

$$n = 2 \quad for\ \beta \ge 1$$

The shock-associated noise spectrum can be expected to exhibit a well-defined peak in the vicinity of

$$f_p = \frac{0.9 M_c c_j}{D_N \beta (1 - M_c \cos\theta)}$$

For a hot jet ($T_j/T_s > 1.1$), the one-third octave band SPL (re 20µPa) is given by:

$$L_P = L_{P\,shock} + \Delta_{shock}$$

where

$$\Delta_{shock} = -15 - 16.1 \log\left(\frac{5.163}{\sigma^{2.55}} + 0.096\sigma^{0.74} \right) + \cdots$$

$$+ 10 \log\left(1 + \frac{17.27}{N_s} \sum_{i=0}^{N_s - 1} C(\sigma)^{i^2} \sum_{j=1}^{N_s - i - 1} \frac{\cos(\sigma q_{ij}) \sin(0.1158 \sigma q_{ij})}{\sigma q_{ij}} \right)$$

and

$$C(\sigma) = 0.8 - 0.2 \log\left(\frac{2.239}{\sigma^{0.2146}} + 0.0987\sigma^{2.75} \right)$$

$$\sigma = 6.91 \beta D_N f / c_2$$

$$N_s = 8 \quad (number\ of\ shocks)$$

$$q_{ij} = (1.7 i c_2 / U_j)\{1 + 0.06[j + \tfrac{1}{2}(i+1)]\}\left[1 - 0.7\left(\frac{U_j}{c_2} \right)\cos\theta \right]$$

Screech

No predictive equations are provided for level of jet screech because it is easily controlled in practice[41]. Screech tones radiate equally in all directions, and the fundamental tone is centered around

$$f_{screech} = \frac{U_c}{L(1 + M_c)}$$

where U_c is the convection velocity of the disturbance in the shear layer, L is the axial length of the first shock cell, and $M_c = U_c / c_j$.

Screech can be virtually eliminated by minor modifications to nozzle design, for example, by the addition of tabs in the exhaust flow, by notching the nozzle perimeter, or by using a non-axisymmetric discharge nozzle.

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8. NOISE-ATTENUATING ELEMENTS

The noise-attenuating properties of pipes, ducts, tanks and vessels, in-line and vent silencers, lagging and propagation outdoors and in rooms is addressed below.

Pipes, ducts, tanks and vessels within which the gas flows act to constrain the sound field through mass and stiffness. Too much acoustic energy within the pipe can have negative consequences however. "Excessive vibration can cause failure or damage to valve and pipe mounted instruments and accessories. Piping cracks, loose flange bolts, and other problems can develop".[42]

The open end of a pipe has a noise attenuating function: some of the sound energy is reflected back from the opening when the wavelength is significantly larger than the pipe exit, the exit is rather abrupt, and high velocity discharge flow is not present.

In-line silencers, if properly selected, are an effective means of reducing noise levels. They accomplish this task by converting acoustic energy into minute amounts of heat energy. Most practical silencers cause a measurable pressure drop. Generally speaking, the more sound that can be attenuated per unit distance, the more pressure drop the silencer develops. One consequence of the pressure drop is flow noise generated within the silencer itself, called *self-noise.* Thus some judgement must be exercised in balancing the competing interests of attenuation and pressure drop.

Lagging of ducts, pipes and vessels attenuate sound as it radiates from pipe and duct walls. The lagging constrains the sound in a sound-absorbing cavity that converts the sound into minute amounts of heat.

8.1. Pipes and Ducts

The noise attenuating performance of pipes and ducts is called in *Transmission Loss*, which is a measure of the ability of the wall to resist transmission of sound. High values of transmission loss correspond to a high degree of sound isolation.

8.1.1. Transmission Loss of Circular Pipes

The Transmission Loss of a circular pipe[43] reaches a minimum at the first mode cut-on frequency of the pipe:

$$TL_{f_0} = 10\log\left[\frac{rt_p^{\,2}}{D_p^{\,3}}\left(\frac{P_2}{P_a}+1\right)\right] + 69.5 + \Delta TL(f, f_0, f_r)\, dB$$

The first mode cut-on frequency f_0 and the ring frequency of the pipe wall f_r are:

$$f_0 = \frac{0.586c}{D_p}$$

$$f_r = \frac{c_L}{\pi D_p}$$

The correction term ΔTL is positive-valued, such that Transmission Loss increases as the frequency moves away from the lowest value at f_0. Strong low frequency attenuation comes about because the pipe walls must be literally stretched by hoop stress in order for the pipe walls to vibrate uniformly (n=0 mode). At f_0, the sound field within the pipe is no longer uniform across the cross-section, allowing other more efficiently radiating pipe wall vibration modes to become active. Above f_r, the radius of curvature of the pipe wall is large compared to a wavelength, and the wall behaves like a flat plate. The Transmission Loss of flat plates increases with frequency.

$$\Delta TL = -20\log\frac{f}{f_0} \text{ for } f \le f_0$$

$$\Delta TL = 13\log\frac{f}{f_0} \text{ for } \mathrm{f}_0 < f \le f_r$$

$$\Delta TL = 20\log\frac{f}{f_0} - 7\log\frac{f_r}{f_0} \text{ for } f > f_r$$

For steel pipe, c_L is 5050 m/sec (16,564 ft/sec) and f_r is $c_L/\pi D_p$.

8.1.2. Transmission Loss of Rectangular Ductwork

The Transmission Loss of rectangular ductwork[43] is equal to that for circular pipe at high frequency, but is typically less at low frequency because of the reduced bending stiffness of the walls. Especially where control of low frequency noise is important, consideration should be given to using circular pipe.

Transmission Loss performance of a duct wall (assuming single as opposed to double layer construction) follows a "mass law", in which mass per unit area and frequency are the only relevant parameters:

$$TL_{duct} = 20\log(f\rho_s) - 45 \text{ (dB)}, f \ge f_{cr}$$

$$TL_{duct} = 13\log\left(\frac{f\rho_s^{\ 2}}{a+b}\right) - 13 \text{ (dB)}, f < f_{cr}$$

$$f_{cr} = \frac{0.520c}{\sqrt{ab}}$$

where ρ_s is mass per unit area in kilogram per square meter and a and b are duct cross-sectional dimensions in meters.

8.1.3. Structural Acoustical Limits

Valve manufacturers recommend limiting control valve noise to 115-120 dB(A) at 1 meter[42,44]. Given that most circular pipe in which control valves are installed has

Transmission Loss on the order of 50 dB, the corresponding interior sound pressure levels is on the order of 165 to 170 dB(A). Indeed, one study[45] indicates that the maximum allowable sound power level to avoid structural failure for pipe with 8 mm wall thickness varies from 170 dB for 10-in. diameter pipe to 160 dB for 36-in. diameter pipe. The function has been parameterized for the purposes of this study as

$$PWL_{Limit} = 185.5 - 1.5\left(\frac{D_p}{1\,\text{in.}}\right) + 0.02\left(\frac{D_p}{1\,\text{in.}}\right)^2$$

$$\text{for } 10\,\text{in.} \le D_p \le 36\,\text{in.}$$

Noise control should be implemented at the source when sound power levels exceeding the structural limit are encountered. Exterior lagging and other "add on" noise control treatments that do not reduce the interior noise level or pipe wall vibration are ineffective.

Note – the structural fatigue criterion given above was developed for petrochemical plants and refineries where continuous operation is usually assumed. In cases of infrequent operation the criterion could probably be relaxed somewhat. The criterion could also probably be relaxed somewhat for pipes with thicker walls. No data is available for either case at this time.

8.2. Acoustical lagging

Acoustical lagging refers to the treatment of piping and equipment to reduce the radiation of noise to surrounding areas. Pipe lagging performance is expressed in terms of *Insertion Loss*; high values indicate a high degree of acoustical isolation.

The sound pressure level L_P after installation may be computed from that before installation as:

$$L_{P,after} = L_{P,before} - IL$$

Lagging is selectable by thickness on the System spreadsheet and in the Flow Noise spreadsheet.

Lagging consists of a layer of flexible, high-density sound absorbing material applied directly to the exterior surface of the pipe. A massive, continuous jacket layer is applied over the absorbing material. The jacket constrains sound within the acoustic cavity where some of the sound energy is converted into heat energy.

At low frequencies the entrapped air in the cavity is stiff (with stiffness proportional to the inverse of the cavity depth) and provides an unattenuated path for vibration to travel directly to the jacket, bypassing the acoustical insulation. The jacket adds very little mass to the system, hence negligible attenuation is achieved under these circumstances. Also, it should be observed that to extend low frequency performance, the cavity depth must be increased.

At high frequencies the air in the cavity is less stiff and sound must travel through the acoustical insulation, which attenuates the wave as it travels, until it reaches the jacket, which reflects it back into the cavity. Significant levels of attenuation are achievable provided that

- the jacket is continuous, and
- no significant rigid paths (such as supports) have been introduced between the pipe wall and the jacket.

The absorbing material is typically glass fiber (2½ to 6 pounds per cubic foot density) or mineral fiber (4 to 8 pounds per cubic foot density). Other materials such as calcium silicate and expanded closed-cell foams are not recommended because they are too rigid. When calcium silicate or closed cell foam are desired for thermal isolation, a thin layer should be used next to the pipe. The acoustical lagging provides good thermal insulation as well.

The jacket material is typically 26 to 28 ga. Steel, 16 to 20 ga. Aluminum, or a barium-loaded vinyl material. A common factor among these is that the surface density is approximately 1.25 pounds per square foot. Lead/aluminum laminate has been used in the past.

If periodic inspection is required, a lace-up style removable/reusable blanket may replace the jacket and perhaps the blanket as well. Note that a removable/reusable blanket is susceptible to degradation with wear and the possibility of gaps developing during re-installation.

8.2.1. Lagging Specification

The acoustical lagging shall consist of 2-in., 4-in., or 6-in. thick mineral fiber placed against the pipe wall plus an external jacket incorporating steel, aluminum and/or loaded vinyl to achieve a 1.25 pound per square foot surface weight. If loaded vinyl is used, it shall be sheathed with an exterior metal jacket. Thermal insulation such as calcium silicate or closed-cell synthetic foams shall not be acceptable substitutes for the cavity fill.

All circumferential joints of the insulation should be staggered and sealed with a non-hardening adhesive. Longitudinal seams and adjoining sections are to be firmly butted together and sealed. All gaps and voids are to be packed with loose insulation. Field cut the acoustic insulation to snugly fit around irregular shapes, elbows, flanges and valves. Jacket seams shall overlap by no less than 2 inches; stainless steel banding shall be applied on 9-10 inch centers (use of screws and rivets alone is not recommended).

Insertion Loss performance of the lagging system shall be no less than given below in Table 13 when measured in accordance with ASTM E1222 "The Laboratory Measurement of the Insertion Loss of Pipe Lagging Systems" or by a field test method acceptable to the purchaser.

Table 13: Insertion Loss Performance of Lagging Systems

	31.5	63	125	250	500	1000	2000	4000	8000
2 in.	1	3	4	6	12	22	23	21	20
4 in.	2	4	5	10	15	27	30	24	20
6 in.	4	7	10	15	25	30	30	22	20

8.3. Radiation and Reflection from the Open End of a Pipe

The process of radiation and reflection of sound from the open end of a pipe is well understood in the absence of mean flow. Reflection of sound is most pronounced when the pipe termination is abrupt (rather than extended by means of a horn). A short bell-mouth is considered abrupt for the purposes of this Guide.

Sound having wavelength greater than the pipe opening diameter reflects back into the pipe; sound having wavelength less than the pipe opening diameter propagates freely out into the environment.

The introduction of mean flow complicates the matter considerably. Consider first the limiting cases. For discharge flow $M_j \geq 1$ "reflected" sound is unable to travel upstream into the pipe, and is convected out into the environment with the flow. Thus, no reflection loss occurs in this case: all of the gas-borne sound in the pipe is radiated. Conversely, for intake flow with $M_j < -1$, no sound within the pipe can reach the plane of the inlet: it is convected back into the pipe with the flow. The reflection loss in the latter case is complete: no sound can be radiated.

In reality, however, sound is generated by a high velocity inlet vent. The most probable sources are inlet debris screens, sharp edges near the opening where the mean flow velocity is not yet sonic, and constrained jet noise downstream of the inlet radiating out through the pipe walls.

The following parametric dependence for the reflection loss (IL) has been deduced from data given in two theoretical and empirical studies[46,47]:

$$IL = -10\log_{10}\left[(1+M)^2(1-r_E)\left(\frac{\rho_a c_a}{\rho_1 c_1}\right)\right]$$

where

$$r_E = e^{-\xi_1(ka)\cdot ka}\left(1-e^{-\xi_2(M)\sqrt{ka}}\right)$$

$$\xi_1(ka) = 0.583 + 0.391(ka)$$

$$\xi_2(M) = \left(\frac{1-M}{7.17M^2+0.43M}\right)$$

where IL denotes the Insertion Loss or reduction in sound power level due to the effect, M is positive for discharge and negative for inlet flows, a is the radius of the pipe opening and k is the acoustic wavenumber $2\pi f/c$. The index 0 and 1 for the density and sonic velocity factors refer to ambient and within the pipe, respectively.

8.4. Silencers

Four basic types of in-line silencers exist: dissipative silencers, reactive silencers, combination silencers and vent silencers. With the exception of the vent silencers, all of these types may be used for in-line service. For the purposes of this Guide, silencers are assigned the generic descriptors D, R, C and V, respectively. Most silencers come in various diameters and sizes to accommodate a variety of flows and performance ranges. Four generic grades of performance are referred to in manufacturers' literature: commercial, semi-residential, residential and critical. These grades refer to increasing degrees of performance associated with the named applications, and are assigned generic descriptors –2, -3, -4, and -5. More detailed information on silencers is available from Universal[48] and Burgess-Manning[49].

Silencer performance is expressed in terms of *DIL (Dynamic Insertion Loss)* which is the Insertion Loss under actual service conditions of flow, temperature, etc.

Actual silencer DIL performance is strongly affected by a number of parameters. Silencer performance figures tabulated below are generic and are for preliminary design purposes only. Silencer performance figures for the actual service conditions anticipated should be requested from silencer manufacturers.

For silencer conditions exceeding 15 psig pressure and 20 in. Hg vacuum, ASME Code construction (Section VIII, Div. 1) is typically recommended. It should be noted that higher temperatures alter the effective properties of the acoustic fill in dissipative silencers and require larger volumes for reactive silencers. Acoustical absorption materials are typically rated for temperatures not exceeding 325 °F, while the silencer bodies are typically rated for 500 °F. Make sure that the absorbing fill is rated for the entire range of expected flow temperatures.

The in-line silencers typically associated with control valves have special design considerations and are not addressed here.

8.4.1. Dissipative Silencers

Dissipative silencers attenuate sound by placing sound absorptive materials in contact with the flow. They tend to perform better at higher frequencies. Increased length, greater depth of fill, and narrow flow passages contribute to improved acoustical performance. The flow resistance of the acoustical fill must be carefully controlled to ensure optimum performance. Their performance can be degraded by the presence of oil, dust or other contaminants. Fill erosion can also be a problem above 6000 feet per minute. In such cases, fill protection can be improved, but at the expense of high frequency performance. High performance dissipative silencers have a pressure drop approximately equal to one velocity pressure head ($K=1$).

Dissipative silencers are ideal for axial compressor inlets, fans and blowers, some very low pressure vents (< 15 psig) and other applications where primarily high frequency sound (above 500 Hz) is to be attenuated and low pressure drop is required.

Dissipative silencers are constructed in several configurations. Generic designations have been assigned to the silencer types to facilitate incorporation into the workbook.

- Concentric (DC): sound-absorbing material in a recessed cavity behind perforated metal. The flow path is straight with no restrictions. This type of silencer produces very little pressure drop, but must be many inlet duct diameters long to achieve moderate levels of performance. Two silencer types are documented: DC-2 and DC-4, which refer to commercial and residential grade concentric dissipative silencers.
- Annular (DA): sound-absorbing material is located behind perforated walls and within a streamlined, sound-absorbing centerbody. The flow path is altered by the presence of the centerbody, hence pressure drop is greater than for a concentric silencer. Higher DIL performance is possible in a more compact package. Three annular types are documented: DA-3, DA-4 and DA-5, corresponding to typical semi-residential, residential and critical grade silencers.
- Splitter (DS): sound-absorbing material is located in streamlined, parallel baffles placed in the flow. Performance is controlled by the percent open area (POA), the splitter depth and the ratio of length to splitter gap width. DS-25, DS-33 and DS-50 correspond to splitter silencers with 25%, 33% and 50% open area respectively.
- Tubular (DT): sound-absorbing material is packed into a volume that is traversed by a number of perforated parallel tubes that carry the flow. Similar DIL performance as a splitter silencer can be obtained in about 2/3 the length, but with increased pressure drop. DT-33-1, –2 and –3 refer to three lengths of this type of silencer.

Typical dissipative silencer performance is tabulated below in Table 14, along with the pressure loss factor *K*, typical length to silencer diameter (*L/D)* and length to inlet pipe diameter (*L/P*) ratios, and typical percent open area figures.

Silencer Pressure Drop can be estimated from

$$\Delta P = K \times \tfrac{1}{2}\,\rho U^2$$

Table 14: Typical Dissipative Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
DC-2	3	4	7	14	20	20	16	10	0.25	4.5	6.4	95
DC-4	5	10	20	30	40	45	40	35	0.25	5.0	13.0	95
DA-3	5	7	11	22	32	32	28	22	0.85	2.0	2.3	50
DA-4	5	8	14	24	34	36	32	26	0.85	2.8	3.7	50
DA-5	5	11	20	30	40	43	40	35	0.75	2.2	4.4	50
DS-50	10	22	30	35	38	34	23	15	0.60	4.0	4.0	50
DS-33	10	25	35	45	50	50	45	35	0.70	4.0	4.0	33
DS-25	10	26	40	55	60	63	60	50	0.90	4.0	4.0	25
DT-33-1	7	9	12	17	21	22	20	17	0.80	1.0	1.0	33
DT-33-2	10	16	22	33	42	44	41	36	0.90	2.0	2.0	33
DT-33-3	12	20	30	45	58	60	57	50	1.00	3.0	3.0	33

Dissipative silencer DIL decreases with increasing discharge velocity (where the sound travels with the flow). Conversely, the DIL increases on an intake system.

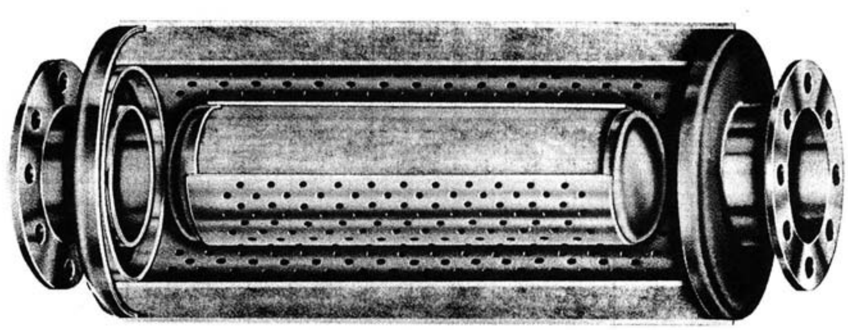

Figure 10: Dissipative Silencer

(Burgess-Manning[49])

8.4.2. Reactive Silencers

Reactive silencers attenuate sound by presenting an acoustical impedance that reduces passage of the acoustic wave. This is accomplished by using one or more chambers connected by tubes. They tend to perform better at low frequencies. Increased volume and number of chambers contribute to improved acoustical performance. High performance reactive silencers have a pressure drop approximately equal to four velocity pressure heads ($K = 4$). Lower pressure drop configurations are available, but performance is reduced.

Reactive silencers are appropriate for rotary lobe and reciprocating compressors, and any application where low frequency noise is to be attenuated and significant pressure drop can be tolerated.

Reactive silencers tabulated below cover low (L) and high (H) pressure drop ranges and all four generic grades of performance –2 through –5.

Table 15: Typical Reactive Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
R-2-L	10	20	28	22	15	13	10	8	0.5	3.0	7.1	50
R-2-H	12	20	27	23	18	17	16	15	4.2	3.0	7.0	50
R-3-L	16	28	35	28	20	15	12	10	1.0	3.7	9.8	50
R-3-H	16	25	33	27	23	20	20	20	4.6	3.4	8.0	50
R-4-H	20	30	35	30	27	25	24	24	5.0	3.7	9.8	50
P–5–H	25	35	36	35	32	29	28	28	5.3	4.4	11.8	50

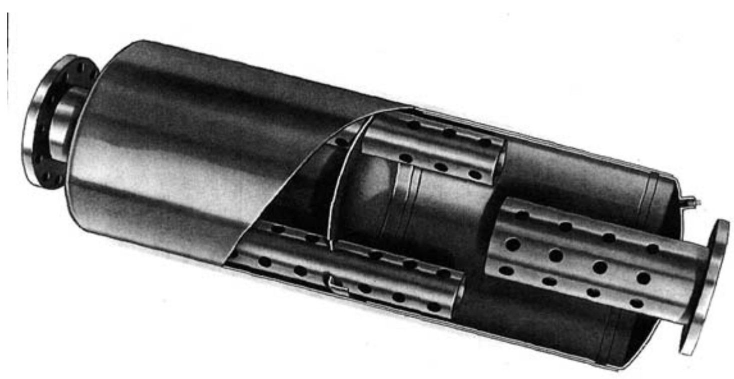

Figure 11: Reactive Silencer

(Universal Silencer[48])

8.4.3. Combination Silencers

Combination silencers attenuate sound using one or more elements of each of the dissipative and reactive type to achieve an insertion loss spectrum combining the benefits of both types. Combination silencers are often used on rotary lobe blowers and compressors.

Combinations tabulated below include: DCR, a short dissipative concentric silencer followed by a reactive chamber, VDR, a diffuser basket followed by a lined reactive chamber, VDA, a diffuser basket followed by a simple dissipative annular silencer, and three grades of VDC, a diffuser basket followed by a dissipative concentric silencer.

Table 16: Typical Combination Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
DCR	12	20	30	35	35	22	20	15	1	2.60	6.00	50
VDR	21	25	29	35	38	38	37	34	13	4.80	17.00	50
VDA	12	21	23	25	34	42	44	43	10	2.25	7.50	50
VDC-3	15	22	30	36	39	38	35	25	10	5.30	13.25	50
VDC-4	19	28	38	43	44	48	57	50	20	7.00	17.50	50
VDC-5	20	40	53	55	53	59	65	61	30	8.60	21.50	50

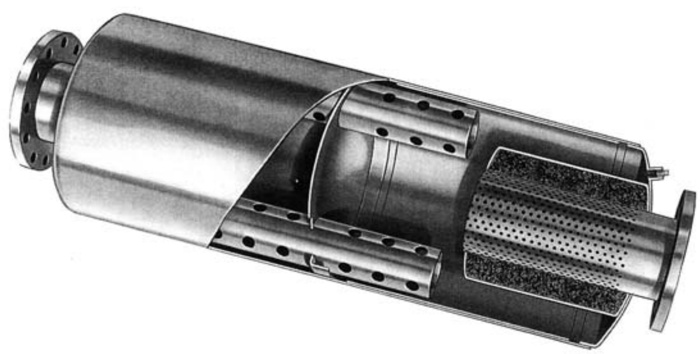

Figure 12: Combination Silencer

(Universal Silencer[48])

8.4.4. Vent Silencers

Vent silencers are a special type of dissipative silencer used to reduce noise from high velocity gas discharges. The vent silencer consists of one or more diffuser baskets that break the discharge jet into a number of smaller jets. This has the effect of shifting the peak frequency f_p upward by several octaves. A dissipative splitter silencer follows the basket. With the peak frequency shifted, the splitter silencer can achieve high *DIL* performance in a short distance. Vent silencers have a pressure drop approximately equal to ten velocity pressure heads per diffuser basket.

Note that sounds already present in the gas flow before reaching the outlet are not frequency-shifted by the diffuser basket. Some of the sound is reflected back into the pipe. In any event, attenuation of low-frequency sound energy in the flow should not be expected to be as dramatic as for the jet mixing noise: in this case, the vent silencer functions as a simple dissipative silencer.

The designation 2VS used below refers to two diffuser baskets applied in series. Four grades each of type VS and 2VS are documented.

Dynamic Insertion Loss performance of silencers for gas vent applications is tabulated below:

Table 17: Typical Vent Silencer DIL Performance

	63	125	250	500	1000	2000	4000	8000	K	L/D	L/P	POA
VS-2	7	6	9	14	19	21	20	19	11.75	2.0	5.1	50
VS-3	10	11	16	25	31	33	32	30	12.00	2.7	6.8	50
VS-4	13	17	24	36	44	46	43	40	12.25	3.4	8.5	50
VS-5	17	22	32	47	56	58	56	50	12.50	4.1	10.2	50
2VS-2	12	10	13	17	21	23	22	21	20.60	2.0	5.1	50
2VS-3	15	15	20	28	33	35	34	32	21.00	2.7	6.8	50
2VS-4	18	21	28	39	46	48	45	42	21.40	3.4	8.5	50
2VS-5	22	26	36	50	58	60	58	52	21.90	4.1	10.2	50

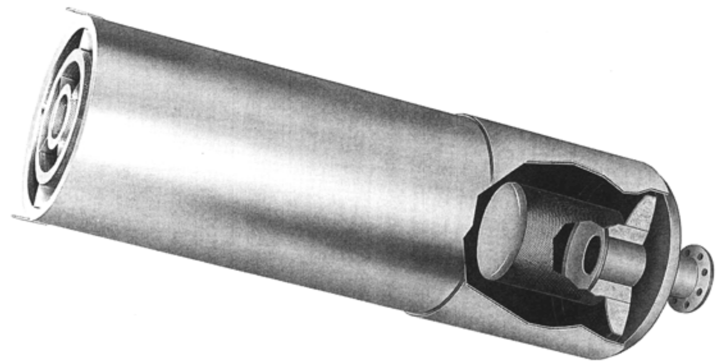

Figure 13: Vent Silencer

(Universal Silencer[48])

8.4.5. Silencer Self-Noise

The term self-noise in relation to a silencer refers to noise generated by the flow of air through the silencer. If poorly selected, the flow noise can severely impact the net performance of the silencer. The silencer self-noise in octave bands can be estimated (after Beranek and Ver**Error! Bookmark not defined.**) as:

$$L_W = -145 + 55\log_{10}\left(\frac{U}{\text{ft/min}}\right) + 10\log_{10}\left(\frac{A_F}{\text{ft}^2}\right) - 45\log_{10}\left(\frac{POA}{100}\right) - 25\log_{10}\left(\frac{460 + T(°F)}{530(°F)}\right)$$

An extra term has been added to L_W to account for gases other than air:

$$L_W' = L_W + 10\log_{10}\left(\frac{MW}{28.967}\right)$$

This sound power level is added to the sound power level leaving the silencer on the quieter side.

In Vér's analysis, the self-noise is presented as constant across all octave bands. To account for the experience of others[48] the following *ad hoc* corrections are recommended:

Table 18: Proposed Octave Band Corrections for Silencer Self-Noise

	31.5	63	125	250	500	1000	2000	4000	8000
L_W corrections	+15	+10	+5	0	0	0	0	0	0

[42] Masoneilan Dresser "Noise Control Manual", Bulletin OZ3000, April 1995.

[43] David A. Bies and Colin H. Hansen, *Engineering Noise Control: Theory and Practice, Second Edition"*, E & FN Spon, London 1996

[44] *Fisher-Rosemount Valve and Actuator Catalogs*, Fisher Controls International, Inc., 1997

[45] V. A. Carucci and R. T. Mueller, "Acoustically Induced Piping Vibration in High Capacity Pressure Reducing Systems", Paper No. 82-WA/PVP-8, American Society of Mechanical Engineers, New York, 1982.

[46] F. von Mechel, D. Schilz and J. Dietz, "Akustische Imedanz einer Luftdurchströmten Öffnung", *Akustika* **15,** 199-206, 1965.

[47] P.O.A.L. Davies, "Realistic Models for Predicting Sound Propagation in Flow Duct Systems", *Noise Control Engineering Journal*, **40** (1), 135-142, Jan-Feb 1993.

[48] Bill G. Golden, Jim R. Cummins jr., "Silencer Application Handbook" , Universal Silencer, Stoughton, Wisconsin, 1993

[49] "Industrial Silencing Handbook", Burgess-Manning, Inc., Orchard Park NY, 1985

9. WORKBOOK EXAMPLES

9.1. Example No. 1: Nitrogen Venting

In the first example, we assume that nitrogen is to be vented after a low-temperature wind tunnel experiment (described in Example No.3). Our attention will be focused in this example on noise from the gas vent and flow noise from the gas rushing through the piping on the way to the vent.

First, we evaluate the criteria based on the methods of the *Specifications Guide*.

Noise Criteria:	**Gas Vent**	
	Group 2:	85 dB(A) Baseline for Group 2
	Adjustments:	+5 dB(A) Remote Location
	Adjustments:	+5 dB(A) Infrequent Operation
	MPSL:	95 dB(A) @ 1 meter
	Outdoor PWL:	Applicable

	Outdoor Piping to Vent	
	Group 3:	80 dB(A) Baseline for Group 3
	Adjustments:	+5 dB(A) Remote Location
	Adjustments:	+5 dB(A) Infrequent Operation
	MPSL:	90 dB(A) @ 1 meter
	Outdoor PWL:	Applicable

Next, we evaluate component noise emission beginning with the Gas Vent. Assume that after coming to rest the nitrogen gas pressure is 600 kPa (8.5 atmospheres) and the temperature is 115 °K (200 °R). Pressure and temperature at the exit are assumed to be 1 atmosphere and 100°F. The valve and downstream pipe diameter are assumed to both be 400 mm (16 inches), and the remote observation position is 137 meters (450 ft.) away. Since no silencer is yet present, the silencer diameter is entered as that of the discharge pipe. The vent discharges skyward, so the observation angle relative to the opening is greater than 90°.

This data is entered in the appropriate cells in the "Gas Vents and Reliefs Spreadsheet". Without a silencer, the sound pressure level at 1 meter is 140 dB(A) and 98 dB(A) at 137 meters! A vent silencer is clearly required.

A preliminary silencer selection is made on the Silencers Spreadsheet. Some knowledge of the flow conditions downstream of the valve is required. The first input is the gas volume flow (actual volume) 224 m^3/sec that was computed on the Gas Vents and Reliefs Spreadsheet. Next the assumed flow conditions upstream and downstream of the silencer are entered: (guess 600 kPa and 115°K upstream, 100 kPA and 300°K downstream). In this case, a vent silencer is appropriate, and the most aggressive model is chosen (2VS-5). The effective flow diameter (the diameter of the pipe with the same open area as the silencer) should be varied until the warning indicator on the "Silencer Velocity" line (row 26) is no longer highlighted. Note that a hint (Minimum Flow Path Diameter) is given in the row above. The Insertion Loss and Self-Noise computed in Section 3.a and highlighted with the salmon colored background are copied into Section 4.a of the Gas Vents and Reliefs Spreadsheet.

With the silencer "installed", the estimated level at 1 meter is reduced to 104 dB(A). At the remote location, the estimated level is 61 dB(A). Note that both the A-weighted sound pressure level and sound power level output are higher than permitted by the *Specifications Guide*. Note also that much of the noise at higher frequencies is actually a consequence of self-noise. Thus, it appears that a silencer with a still larger flow area would have been more beneficial.

Finally, the flow noise estimation is performed. With the gas "Nitrogen" selected, the mass flow (once again calculated on the Gas Vents and Reliefs Spreadsheet) and flow parameters are entered. The pipe diameter, wall thickness and length are entered next. Finally, the piping complexity is computed based on components present in the piping system: we assume that the pipe has one welded 90° turn in a 100 ft. length.

The estimated sound level 1 meter from the pipe is 100 dB(A) in the absence of lagging (see Section 6.a). Also, the outdoor sound power level limit is exceeded. In Section 6.b a 4 inch thickness of lagging is selected, which brings the radiated flow noise down to a more bearable 89 dB(A) at 1 meter. Radiated sound power is expected to be only slightly above the maximum permissible emission in two octave bands.

The relevant Spreadsheets are copied onto the following six pages. This concludes the discussion of Example No. 1.

GAS VENTS AND RELIEFS
All Gases Except Steam

1. Initial Upstream Gas Conditions

1a. Select Gas

1.a.1 Gas	Nitrogen (N2)		
MW	28.016	[mass/mole]	
γ	1.398	[1]	
R	55.15	[(ft lbf)/(lbm °R)]	

1b. Enter Reservoir Initial Pressure and Temperature

1.b.1 Reservoir Pressure	**600**	[kPa]	P_1	
1.b. 2 Reservoir Temperature	**115**	[° K]	T_1	
1.b.3 Reservoir Volume (optional)	***86500***	[ft^3]	V	*(for use in fixed volume blowdown applications)*

1c. Calculate Reservoir Initial Density

Reservoir Density	1.114	[lb/cu ft]	ρ_0

2. Downstream Gas Conditions, assuming isentropic flow

2a. Enter Downstream Conditions

Exit Pressure	**1**	[atm]	P_2
Exit Temperature	**20**	[° C]	T_2

2b. Calculate Jet Flow Parameters

Exit Density	0.074	[lb/cu ft]	ρ_2
Stream Mach Number	1.82	[1]	M_j
Stream Density	5.001	[kg/cu m]	ρ_j
		[° K]	
		[ft/sec]	
		[ft/sec]	

	[in]
	[ft^2]
	[in]
	[in]
	[ft]

3.a.6 Observation Angle re Axis of Opening		> 90°	θ

3b. Calculate Blowdown Parameters

Initial Flow Rate	7,228	[cfs]	scfm
Initial Flow Rate	439,665	[cfm]	acfm
Initial Mass Flow	542	[lb/sec]	
Blowdown Time	341.26	[sec]	

4. Estimated Noise Emission, Silenced

4a. Estimated Silenced Sound Power Level (L_W)

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
L_W, Vent	126	133	140	145	147	146	142	135	128	**150**
IL, Silencer [from Mfr or Silencer Sheet]	**9**	**17**	**22**	**32**	**47**	**56**	**58**	**56**	**50**	
L_W, Vent, Silenced	118	116	118	113	100	90	84	79	78	**107**
L_W, Silencer Self Noise	**90**	**122**	**117**	**112**	**112**	**112**	**112**	**112**	**112**	
L_W, Total	118	123	120	116	112	112	112	112	112	**119**
Directivity, Silencer Outlet	0	0	0	0	0	-1	-3	-7	-13	
Directional L_W, Silenced	118	**123**	**120**	**116**	**112**	**111**	**109**	**105**	99	**116**
Maximum Permissible Outdoor Sound Power Levels		*127*	*120*	*113*	*110*	*108*	*107*	*107*	*106*	

4b. Estimated Silenced Sound Pressure Level (L_P) at Observation Position

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
Directional L_W, Silenced	118	123	120	116	112	111	109	105	99	**116**
Geometric Divergence to Obs. Position	-51	-51	-51	-51	-51	-51	-51	-51	-51	
Lp, Silenced, at 450 ft.	67	72	70	65	62	60	58	54	48	66
A-weighted Sound Pressure Level Target										**85**

4c. Estimated Silenced Sound Pressure Level (L_P) at 1 meter

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
L_W, Total	118	123	120	116	112	112	112	112	112	**119**
Directivity, 90°	0	0	0	0	0	-1	-3	-7	-13	
Geometric Divergence to 1 meter	-8	-8	-8	-8	-8	-8	-8	-8	-8	
L_P, Silenced, at 1 meter	110	115	**112**	**107**	**104**	**103**	**101**	**97**	91	**108**
Maximum Permissible Sound Level (MPSL) for Gas Vent										**95**

5. Estimated Overall Noise Emission, Unsilenced

5a. Estimated Overall Sound Power Level

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
L_W, Vent	126	133	140	145	147	146	142	135	128	**150**

PRELIMINARY SILENCER SELECTION WORKSHEET

1. Enter Flow Conditions

1.a. Select Gas	Nitrogen (N2)		
Gas Mol. Weight	28.02	[1]	*MW*
γ	1.398	[1]	
R	55.15	[(ft lbf)/(lbm °R)]	
1.b. Gas Volume Flow (acfm)	**7200**	[cfs]	Q
1.c. Approx. Inlet Pressure (after Valve)	**600**	[kPa]	P_1
1.d. Stream Temperature	**115**	[° K]	T_j
1.e. Downstream Ambient Pressure	**1.0**	[atm]	P_a
1.f. Downstream Ambient Temperature	**20**	[° C]	T_a
1.g. Downstream Ambient Density	0.074	[lb/cu ft]	ρ_α
1.h. Sonic Velocity	348.9	[m/sec]	c_j

2. Select Silencer

2.a. Silencer Type	Vent			
2.b. Silencer Selection (see Section 8)	VS-5			
2.c. Effective Flow Diameter	**96**	[in]		D_f
Minimum Flow Path Diameter	89	[in]		
Silencer Velocity	143	[ft/sec]	Below Design Velocity	
Maximum Design Velocity	167	[ft/sec]		
Silencer Diameter	239	[in]		
Silencer Length	82	[ft]		
Silencer K Value	12.50	[1]		K
Pressure Drop	737.7	[lb/sq ft]		ΔP

3. Estimate Insertion Loss (IL) and Self-Noise

3a. Estimated Silencer Performance Data

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*
Estimated Insertion Loss (dB)	9	17	22	32	47	56	58	56	50
Estimated Self-Noise (L_W dB re 1 pW)	127	122	117	112	112	112	112	112	112

Silencer performance can be affected by many factors, some of which are accounted for only approximately here. Manufacturer's Data should be used wherever available.

4. Silencer Types

Silencers are manufactured in a variety of configurations to accomadate many applications. For the purposes of this Guide various silencer types are designated by letters and a number. The letters indicate the components of the silencer: Dissipative (D), Reactive (R), Vent (V) with corresponding subtypes. Some silencers combine aspects of each of these basic types. The number designation corresponds to a generic grade of performance: (2) is Commercial, (3) is Semi-Residential, (4) is Residential, and (5) is Critical grade. Refer to the Manual for guidance in the selection of appropriate silencer types for your application.

	Dissipative				React.	Vent		Note
Silencer Type	Concentric	Annular	Splitter	Tube	Chambers	Single Diffuser	Dual Diffuser	Grade, Description
DC-2	×							Commercial
DC-4	×							Residential
DA-3		×						Semi-Residnt'l
DA-4		×						Residential
DA-5		×						Critical
DS-50			×					50% Open
DS-33			×					33% Open
DS-25			×					25% Open
DT-33-1				×				Short
DT-33-2				×				Medium
DT-33-3				×				Long
R-2-L					×			Low ΔP
R-2-H					×			High ΔP
R-3-L					×			Low ΔP
R-3-H					×			High ΔP
R-4-H					×			High ΔP
R-5-H					×			High ΔP
DCR	×				×			
VDR	×				×	×		
VDA		×			×	×		
VDC-3	×					×		Semi-Residnt'l
VDC-4	×					×		Residential
VDC-5	×					×		Critical
VS-2						×		Commercial
VS-3						×		Semi-Residnt'l
VS-4						×		Residential
VS-5						×		Critical
2VS-2							×	Commercial
2VS-3							×	Semi-Residnt'l
2VS-4							×	Residential

FLOW NOISE IN PIPES

OVERVIEW

Flow noise is a special case of noise emission because it arises from the interaction of the turbulent boundary layer in the gas with the pipe walls and is therefore generated throughout the system. This Spreadsheet provides computations for use in evaluating a single length of piping or in evaluating a System. This Spreadsheet performs computations of Sound Power Level (L_W) and Sound Pressure Level (L_P) at 1 meter for piping with and without acoustical lagging.

Noise emission data for use in the Integrated System Analysis is presented in Line 6c and corresponds to radiation from an unlagged pipe 10 feet in length. The effects of lagging and actual pipe length are accounted for by selections made in the System Input-Output Spreadsheet. Note however that there is no automatic "feedback" from the System Input-Output Spreadsheet regarding other important parameters such as gas flow conditions and pipe dimensions and thickness. Those inputs must be made manually in this Spreadsheet in order for the computation to be correct.

Computations are based on the number of components in a 10-ft length of pipe. The most correct way to perform this computation is to obtain noise emission estimates for each 10-ft. length and then sum the results (on an energy basis) as shown in the Calculator Spreadsheet, Section 7. An approximate method is to use as inputs the total number of each component in the piping system divided by the number of 10-ft lengths in the system.

1. Select Gas

1a. Gas	Nitrogen (N2)		
Molecular Weight	28.02	[mass/mole]	*MW*
Ratio of Specific Heats	1.398	[1]	γ
Gas Constant	55.15	[(ft lbf)/(lbm °R)]	*R*

2. Flow Parameters

2a. Mass Flow Through Pipe	**224**	[kg/sec]	*m'*
2b. Interior Total Pressure	**600**	[kPa]	*P*
2c. Gas Flow Total Temperature	**115**	[° K]	*T*

3. Piping Dimensions

3a. Pipe Inside Diameter	**16.0**	[in]	D_p
3b. Pipe Wall Thickness	**0.310**	[in]	t_p
3c.Length of Pipe	**100**	[ft]	*L*

4. Piping Complexity

4a. Enter the total number of listed components used in the piping section under consideration.

This information is used to determine K, the number of velocity heads per 10 ft (3 m) of piping.

Straight Pipe		1				
45° Elbow	Screwed	**0**	Welded, R/D=1	**0**	Welded, R/D=1.5	**0**
90° Elbow	Screwed	**0**	Welded, R/D=1	**1**	Welded, R/D=1.5	**0**
180° Elbow	Screwed	**0**	Welded, R/D=1	**0**	Welded, R/D=1.5	**0**
Tees (Screwed)	Thru Branch	**0**	Through Run	**0**		
Tees (Welded)	Thru Branch	**0**	Through Run	**0**		

Reducer	D_2/D_1= 0.3	**0**	D_2/D_1= 0.5	**0**	D_2/D_1= 0.7	**0**
Expander	D_2/D_1= 3	**0**	D_2/D_1= 2	**0**	D_2/D_1= 1.25	**0**
Sudden Contraction	D_2/D_1= 0.1	**0**	D_2/D_1 = 0.33	**0**	D_2/D_1 = .80	**0**
Sudden Expansion	D_2/D_1= 10	**0**	D_2/D_1= 3	**0**	D_2/D_1= 1.25	**0**

Total "K" factor 0.17 [1]

5. Flow Parameters (calculated)

Density	1.114	[lb/cu ft]		ρ
Face area of pipe	1.396	[ft^2]		A_p
Flow Velocity	317.5	[ft/sec]		U
Sonic Velocity	218.8	[m/sec]		c_2
Mach Number	0.442	[1]	*Warning: Velocity > 0.3 M*	M
Ring Frequency of Pipe	4050	[Hz]		f_r
Jet Spectrum Peak Frequency	47.62	[Hz]		f_p
1st Mode Pipe Cutoff Frequency	315.5	[Hz]		f_c

6. Estimated Noise Emission

6a. Estimated Noise Emission with No Lagging

Sound Pressure Levels (L_P), 1 m	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	***A***
L_W Radiated from Pipe per 10 ft.	118	116	115	113	109	98	86	82	62	**109**
Geometric Divergence to 1 meter	-10	-10	-10	-10	-10	-10	-10	-10	-10	
L_P at 1 meter	108	106	**105**	**103**	**99**	**88**	77	72	52	**99**
Maximum Permissible Sound Level (MPSL) for Pipe-Radiated Flow Noise										**90**

Sound Power Levels (L_W)	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	***A***
L_W Radiated from Pipe per 10 ft.	118	116	115	113	109	98	86	82	62	**109**
Correction from 10 ft. to Full Length	10	10	10	10	10	10	10	10	10	
L_W Radiated from Full Length	128	**126**	**125**	**123**	**119**	**108**	96	92	72	**119**
Maximum Permissible Outdoor Sound Power Levels		*127*	*120*	*113*	*110*	*108*	*107*	*107*	*106*	

6b. Estimated Noise Emission with Lagging

Sound Pressure Levels (L_P), 1 m	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	***A***
L_W per 10 ft. Length, Unlagged	118	116	115	113	109	98	86	82	62	**109**
Lagging IL 2 in.	-1	-3	-4	-6	-12	-22	-23	-21	-20	
L_W per 10 ft. Length, Lagged	117	113	111	107	97	76	63	61	42	**101**
Geometric Divergence to 1 meter	-10	-10	-10	-10	-10	-10	-10	-10	-10	
L_P at 1 meter	107	103	101	**97**	87	66	54	51	32	**91**
Maximum Permissible Sound Level (MPSL) for Pipe-Radiated Flow Noise										**90**

Sound Power Levels (L_W)	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	***A***
L_W Radiated from Full Length	128	126	125	123	119	108	96	92	72	**119**
Lagging IL (from above)	-1	-3	-4	-6	-12	-22	-23	-21	-20	
L_W, Full Length, Lagged	127	**123**	**121**	**117**	**107**	86	73	71	52	**111**
Maximum Permissible Outdoor Sound Power Levels		*127*	*120*	*113*	*110*	*108*	*107*	*107*	*106*	

9.2. Example No. 2: Control Valve

The control valve from the nitrogen venting system of Section 9.1 is considered. The first order of business is to determine the noise emission criterion for the valve according to the *Specifications Guide*.

Control Valve	
Group 1:	85 dB(A) Baseline
Adjustments:	+5 dB(A) Remote
Adjustments:	+5 dB(A) Infrequent
MPSL:	95 dB(A) @ 1 meter
Outdoors PWL:	Applicable

We will assume that the mass flow of 224 kg/sec passes through one butterfly control valve that is 60° open. The gas pressures upstream and downstream of the valve are assumed to be 900 kPA and 600 kPA, respectively.

An crude valve selection is made by means of an iterative process in which the selected valve C_V, diameter D_V and open angle are adjusted until they are similar to the approximate C_V and diameter required.

> *Note - It turns out that the sound power level inside the pipe due to the control valve operation exceeds the structural fatigue criterion. This situation could be alleviated by addition of a pressure-reducing plate downstream of the valve. Also note that the fatigue criterion was developed in relation to petrochemical plants and refineries, where operation is more or less continuous. Infrequent operation may provide some leeway here.*

Section 5.a of the Spreadsheet shows that the control valve sound pressure level at 1 meter from the pipe wall is estimated at 130 dB(A) with no noise control treatments. Section 5.b permits the ad hoc addition of noise control options. Addition of a downstream resistance plate brings the SPL down to 115 dB(A), which condition is marginally acceptable from a fatigue standpoint, but not yet acceptable from a hearing conservation standpoint. The further selection of a downstream in-line silencer brings the SPL down to a more bearable 105 dB(A). Further noise control options include addition of valve trim (if available for this valve), an upstream in-line silencer, or external pipe lagging (not addressed on this Spreadsheet).

The Control Valve Spreadsheet is reproduced on the following two pages. This concludes the discussion of Example No.2.

CONTROL VALVE NOISE ESTIMATION

1. Select Flow Conditions

1a. Gas	Nitrogen (N2)		
Specific Gravity	0.97	[1]	G
Ratio of Specific Heats	1.40	[1]	γ
1b. Gas Compressibility Factor	**1**	[1]	Z
1c. Mass Flow	**224**	[kg/sec]	m'
1d. Upstream Pressure	**900**	[kPa]	P_1
1e. Upstream Temperature	**115**	[° K]	T_1
1f. Downstream Pressure	**600**	[kPa]	P_2

2. Select a Candidate Valve Type, Perform Approximate Sizing

2a. Select Valve Type

Type: Butterfly valve, swing-through vane, Flow To: N/A, Travel: 60º open

Flow is	Sonic			
Approx. C_V required	4067		C_V	(Iterate with Line 3.a)
Approx. D_V required	14.5	[in]	D_V	(Iterate with Line 3.b-3.d)
Approximate C_V Wide Open	11297		C_V	

3. Make Valve and Pipe Selection

3a. Select C_V	**4100**	[Cv]	C_V
3b. Valve Nominal Diameter	**16**	[in]	D_V
3c. Pipe Diam. Downstream	**16**	[in]	D_D
3d. Pipe Diam. Upstream	**16**	[in]	D_U
3e. Pipe Thickness	**0.31**	[in]	t_p

4. Relevant Acoustical Parameters

Jet Peak Frequency	537	[Hz]	f_p
Pipe First Mode Cut-on Freq.	1515	[Hz]	f_c
Pipe Ring Frequency	3955	[Hz]	f_r

Internal Overall L_W	173	dB L_W	L_W
Structural Limit Overall L_W	167	dB L_W	L_{WS}
Internal PWL - Structural Limit	6	dB L_W	ABOVE STRUCTURAL FATIGUE CRITERION

5. Estimated Noise Emission

5a. Calculate Octave Band Sound Pressure Levels 1m from Pipe

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
Internal L_P	164	167	170	175	178	175	172	168	165	**180**
Pipe TL	81	75	69	63	57	51	49	52	56	
L_g	3	3	3	3	3	3	3	3	3	
L_P at 1 m from Pipe Centerline	85	94	104	**115**	**123**	**126**	**125**	**118**	**111**	**130**
Maximum Permissible Sound Level (MPSL) for Control Valve										**95**

5b. Add the Benefit of Control Valve Noise Control Options

☐ Valve Trim ☐ Downstream Valve Silencer ☐ Upstream Valve Silencer ☐ Downstream Resistance Plate

Sound Pressure Level (L_P) at 1 m	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
L_P at 1 m from Pipe Wall	85	94	104	115	123	126	125	118	111	**130**
Insertion Loss of Selected Noise Control	0	0	0	0	0	0	0	0	0	
L_P 1 m from Pipe Wall, Noise Control	85	94	104	**115**	**123**	**126**	**125**	**118**	**111**	**130**
MPSL for Control Valve										**95**

Sound Power Level (L_W) Radiated	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
External L_W of 10 ft. length of pipe	95	104	114	**125**	**133**	**136**	**135**	**128**	**121**	**140**
Maximum Permissible Outdoor Sound Power Levels		*127*	*120*	*113*	*110*	*108*	*107*	*107*	*106*	

5c. Estimated In-Duct Sound Power Level (L_W)

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*A*
L_W Inside Pipe, Downstream	155	158	161	166	169	166	163	159	156	**171**

A. NOMENCLATURE

a	one cross-sectional dimension of a rectangular duct
A_i	area of pipe for orifice or venturi, measured at inner edge
A_j	area of jet, fully expanded
A_o	area of pipe for orifice or venturi, measured at outer edge
A_s	area of inlet debris screen
b	one cross-sectional dimension of a rectangular duct
B	number of blades, rotors or cylinders
BFI	Blade Frequency Index
ΔBN	differential ISO band number
c	sonic velocity
c_a	sonic velocity, ambient
c_e	sonic velocity, expanded gas
c_j	sonic velocity in jet
C	stator chord
C_1	inlet guide vane chord length
C_2	fan rotor chord length
C_N	nozzle coefficient
C_V	sizing coefficient of valve [gal/min per psia$^{1/2}$]
D	diameter
$D(\theta)$	directivity factor of source
D_D	diameter of downstream piping
D_F	diameter of fan
D_i	inner diameter
D_j	jet diameter, fully expanded
D_N	nozzle diameter
D_o	outer diameter
D_P	pipe diameter
D_T	turbine diameter
D_U	upstream piping diameter
D_V	valve diameter
D_W	wire diameter
f_b	blade passage frequency
f_c	critical frequency (first mode cut-on) of pipe
F_D	valve style modifier
f_i	i-th pipe wall flexural resonance frequency
F_L	pressure recovery coefficient
f_p	peak frequency of noise emission
$f_p{}'$	peak frequency of noise emission
f_r	circular pipe ring frequency
G	specific gravity of gas
H	height of equipment
IL	insertion loss
K	pressure loss coefficient
L	length of equipment
L_P	sound pressure level
ΔL_P	differential sound pressure level
L_W	sound power level
L_{WS}	sound power level for structural fatigue
L_{WSN}	sound power level of silencer self-noise
M	Mach number of flow
$m\cdot$	mass flow rate
$\%m$	percent moisture in steam

M_c	convection Mach number
M_j	jet Mach number
M_T	tip speed Mach number
M_{TR}	relative tip speed Mach number
M_{TRD}	relative tip speed Mach number, design
MW	molecular weight
N	rotational speed [sec^{-1}]
P_0	static pressure at vena contracta
P_1	static pressure upstream
P_2	static pressure downstream
P_A	ambient pressure
POA	percent open area [%]
PSE	peak static efficiency [%]
P_{TS}	total static pressure rise across fan
q	lobe number for rotor-stator interactions
Q	volume flow rate
r	distance to observation point
r'	effective distance from acoustic center of large equipment
R	gas constant
r_E	energy reflection coefficient
$R(f)$	room constant, frequency-dependent
RSS	rotor-stator spacing coefficient
S_1	inlet guide vane-fan spacing
S_2	rotor/stator spacing
S_A	area covered by sound-absorbing materials
SE	static efficiency [%]
S_R	area covered by sound-reflecting materials
t	blowdown time
ΔT	differential temperature
T_1	upstream temperature
T_2	downstream temperature
T_3	combustor inlet temperature
$T_{4,ref}$	combustor outlet/turbine inlet temperature, maximum takeoff power
$T_{5,ref}$	turbine outlet temperature, maximum takeoff power
T_a	ambient temperature
T_j	temperature of fully expanded jet
TL	transmission loss
ΔTL	differential transmission loss
T_s	superheat temperature
t_p	pipe wall thickness
u	velocity of local flow perturbation
U	centerline mean flow velocity
U_c	convection flow velocity
U_j	centerline mean flow velocity of jet, fully expanded
TL	transmission loss
V	blowdown volume
V_{TR}	tip velocity of last stage turbine rotor
W	width of equipment
W_A	acoustic power
W_M	mechanical power
Z	gas compressibility factor

α	sound absorption coefficient
β	shock parameter
δ	cutoff factor
Δ	differential spectral level
Δ_{shock}	differential spectral level, shock-associated noise
γ	ratio of specific heats
η	acoustic conversion efficiency
ρ	gas density
ρ_a	ambient gas density
ρ_e	expanded gas density
ρ_j	jet density, fully expanded
ρ_s	mass per unit area
ρ_W	density of water
σ	frequency-dependent shock parameter
ξ	adjusted pressure loss coefficient
ξ_1	reflection parameter
ξ_2	reflection parameter

B. DEFINITION OF NOISE CONTROL TERMS

A-weighting: An electronic filter system in a sound level meter that emphasizes frequencies most likely to cause hearing damage. Sound pressure level readings obtained using this weighting are referred to as "A-weighted sound pressure level" or simply "sound level" and are written with the abbreviation dB(A) or dBA and pronounced "dee-bee-ay".

acoustical lagging: noise control materials applied to the exterior of noise-radiating surfaces. Usually consist of a flexible layer of fibrous materials several inches thick covered with a massive jacket.

Baseline Criterion: As defined in the *Specifications Guide*, a criterion equipment noise emission level in dB(A) that applies to a *Group* of equipment without reference to siting or operational considerations.

blowdown: Relief of a fixed volume of high-pressure gas to atmosphere or a low-pressure tank from the atmosphere. Usually accompanied by high sound levels.

constrained jet: a high-velocity jet of air that is constrained within a pipe or other vessel. Control valves, orifices, venturis and intake vents all possess constrained jets.

conversion efficiency: efficiency of conversion of mechanical power to acoustical power. Typically increases with velocity and turbulence.

cut-on: condition for propagation of sound in a particular mode. Below the cut-on frequency, sound in the given mode attenuates rapidly with distance. Above the cut-on frequency, sound in the given mode propagates freely.

cutoff: a condition in which it is difficult for discrete tones generated in turbomachinery to propagate through the rotor-stator system to the environment.

decibel: dB- a measure of the amount of energy in an acoustic signal. A change of 10 dB indicates a 10-fold energy increase or decrease; a change of 20 dB corresponds to a 100-fold energy increase or decrease, etc.[v] The mathematical formulation of the dB is as the common logarithm of the ratio of the measured sound pressure to that of a signal that is barely audible. Thus, the decibel has no units, and strictly speaking is not a unit itself. However, it is common to state "the sound pressure level is 80 dB".

Design Guide: Reduced-Noise Gas Flow Design Guide.

diameter, fully developed jet: the diameter the jet has attained at a point where the axial core velocity begins to reduce with distance. Near the exit, the jet diameter increases as air is entrained. Usually several times the exit diameter.

[v] Because of the characteristics of human hearing, a ten-fold energy change corresponds to a two-fold change in perceived loudness; a one-hundred-fold energy change to a four-fold loudness perception change, etc.

dipole source: oscillatory force pair that radiates sound. Analogous to two closely spaced *monopole sources* operating out of phase. Typically associated with the interaction of flows and structures.

direct sound: sound that travels from its source to the observation point in a direct line, without striking reflecting obstacles or room surfaces.

dissipative silencer: a silencer that provides *insertion loss* by dissipating acoustic energy. Sound is converted into minute amounts of heat within the fibrous acoustic fill.

duct mode: A pressure pattern across the duct cross-section that propagates down the duct. Uniform pressure across the duct is called the plane-wave mode and propagates at all frequencies. More complex pressure patterns propagate only above their duct mode *cut-on* frequency.

far field: the sound field farther than a characteristic dimension from its source. Characterized by reduction in level with distance (in the absence of sound-reflecting obstacles).

flow noise: noise generated by fluid flow in the turbulent boundary layer of a pipe

free jet: a discharge of high velocity gas into the atmosphere, unconstrained by surrounding structures such as pipes.

Group Number: As defined in the *Specifications Guide*, a classification for equipment with similar noise emission expectations.

isentropic expansion/contraction: expansion or contraction of gas without the addition of entropy. A gas undergoes isentropic expansion or contraction when it is travels from one set of pressure/temperature/density conditions to another without encountering a shock.

Inlet Debris Screen: a screen placed over an air intake to prevent ingestion of debris, birds, etc.

in-line silencer: a silencer placed within the gas flow.

in-line sound power level: sound power level of gas within the flow, as opposed to radiating from the pipe walls.

Insertion Loss: IL, dB- in each octave band, the amount by which source levels are attenuated by the candidate noise control option. Insertion Loss data expressed in dB(A) should be carefully regarded, as the A-weighted level reduction for a given IL spectrum is a function of the original source spectrum.

intake vent: an opening to atmosphere for vacuum intake.

lagging: see *acoustical lagging*

MPSL: As defined in the *Specifications Guide*, the maximum permissible A-weighted Sound Pressure Level measured 1 meter away from the individual equipment item under consideration.

monopole source: an aerodynamic pulsation that emits sound.

near field: the sound field closer than a characteristic dimension from its source. Characterized by variable levels clustered around a more or less stable mean value.

pressure recovery: The degree of difference between the static pressure downstream and that in the vena contracta at a flow constriction. A pressure difference is accompanied by accelerated flow in the vena contracta relative to downstream velocities. Because acoustic *conversion efficiency* increases with velocity, large pressure differences (high recovery) may mean increased noise.

quadrupole source: a rotating (shear) force pair that radiates sound. Analogous to two closely-spaced dipole sources operating out of phase. Associated with turbulence.

reactive silencer: a silencer that provides *insertion loss* by presenting an acoustic impedance. Sound is reflected from the silencer.

Reflection Loss: dB – the numerical difference in sound power level approaching a pipe opening to that which is actually radiated. The remainder is reflected back into the piping system.

reverberant sound: sound that travels to the observation point via one or more room surfaces.

room constant (R): m^2 – an expression of the sound absorbing capacity of a room. Analogous to the area over which radiated sound power is distributed to give *reverberant sound*.

self-noise: flow noise generated by flow through a silencer.

sound-absorbing materials: materials or surfaces that remove sound energy from a given space. Most sound absorbing materials are lightweight and porous and remove sound energy by converting it to minute quantities of heat. A less obvious but powerful sound absorber is a large extent of open air: sound travelling out of open windows, missing walls and into the open sky does not return.

sound level: A-weighted sound pressure level

sound power: watt – the acoustic power associated with a source.

sound power level: L_W, dB – a decibel expression of the sound power. The reference sound power is 10^{-12} watts.

sound pressure: Pa –oscillatory pressure superimposed over static atmospheric pressure.

sound pressure level: L_P, dB – a decibel expression of the sound pressure. The reference sound pressure is 2×10^{-5} Pa.

sound-reflecting materials: materials that do not remove sound energy but reflect it back into the space. Examples would be concrete block, plaster and the ground.

sound source: an equipment item or a part of an equipment item that emits audible noise.

source dimensions: the length, width and height of a rectangular box fitting over the sound source.

Specifications Guide: NASA Glenn Research Center "Guide to Specification of Equipment Noise Emission Levels"

Transmission Loss: TL, dB- in each octave band, the difference between the incident and transmitted sound power levels for the candidate noise control option. Similar to Insertion Loss, but in some cases introduction of the noise control option actually increases the sound power incident on itself. Transmission Loss data expressed in dB(A) should be carefully regarded, as the A-weighted level reduction for a given IL spectrum is a function of the original source spectrum.

vacuum vent: an opening to atmosphere for vacuum intake.

valve trim: a class of devices added within the body of a control valve to reduce noise emission by increasing the peak noise frequency and causing the pressure reduction to occur in smaller steps.

vena contracta: the point of smallest cross-sectional area downstream of a flow constriction.

wave divergence: the numerical difference between the sound power level of a source and the sound pressure recorded at a particular location, absent the effects of directivity. Represents the extent over which sound power must spread itself in a given environment. Because the magnitude of sound pressure power is dimensionally related to the sound power per unit area, spreading the sound power over a large area produces lower sound pressure levels.

C. DECIBEL MATHEMATICS

C.1. Energy Addition

The sound pressure level of a combination of sounds is computed on the assumption that the sounds are uncorrelated. This form of the equation is appropriate for use with calculators and computers.

$$L_{P,total} = 10\log_{10}\left(\sum_i 10^{0.1L_{P,i}}\right)$$

C.2. Energy Subtraction

Sound pressure levels can be subtracted as well. This might be done when attempting to subtract the influence of one machine from a reading on a group, or when attempting to remove the influence of ambient noise from a measurement.

$$L_{P,diff} = 10\log_{10}\left(10^{0.1L_{P,1}} - 10^{0.1L_{P,2}}\right)$$

This computation assumes that the sounds are uncorrelated. This form of the equation is appropriate for use with calculators and computers.

C.3. Mnemomic Method for Addition

A simplified method, accurate to approximately 1 dB, is well suited for spontaneous "in the head" calculations.

Decibel values are added two at a time. When adding a series of numbers, always begin with the lowest value and proceed to the highest. In each case, the sum of the two values will be the value of the greater of the two, plus a factor that depends on the difference between them.

$$L_{P,1+2} = L_{P,1} + \Delta\left(L_{P,1} - L_{P,2}\right)$$

where $L_{P,1} > L_{P,2}$

Δ	$L_{P,1}$ - $L_{P,2}$
+3 dB	0 or 1 dB
+2 dB	2 or 3 dB
+1 dB	4 to 9 dB
+0 dB	10 dB or more

C.4. Calculation of A-weighted levels from octave bands.

A-weighted sound pressure levels or sound power levels may be computed to a reasonable accuracy from an octave band spectrum as follows:

$$L_{PA} = 10\log_{10}\sum_{i} 10^{0.1\left(L_{P,i}+A_i\right)}$$

where the values A_i for the octave bands are given in Table 19.

Table 19: A-weighting Corrections

Octave Band Center Frequency [Hz]	A-weighting Correction [dB]
31.5	-39.4
63	-26.2
125	-16.1
250	-8.6
500	-3.2
1000	+0.0
2000	+1.2
4000	+1.0
8000	-1.1